ABSTRACT

PAPROCKI, DANIEL. A Quantitative Analysis to Determine Methods to Improve an Industrial Compressed Air System. (Under the direction of Dr. Stephen Terry).

Compressed air systems are an integral part of many manufacturing facilities. Compressed air is used for many things, including pneumatic controls, manufacturing equipment, air motors, air tools, and blow off nozzles. Despite the versatility of compressed air, it comes at a great cost, as approximately 80 percent of the energy used by the air compressor is rejected as heat. The remaining 20 percent of the energy is converted to compressed air. Thus, it is important to produce and use compressed air as efficiently as possible, which controls can help accomplish.

The subject facility has a compressed air system that consists of four rotary screw type compressors. Currently this system of compressors is not using system master controls, which would help sequence the compressors according to compressed air demand.

After collecting amperage and pressure readings through data logging, several analyses were carried out. First, a method to estimate power factor was created, then compressor power for each compressor was calculated using amperage, power factor, and compressor motor efficiency. Additionally, a technique to calculate volumetric flow using compressor power was developed. The estimation of power factor and volumetric flow are essential when determining compressor power and analyzing compressed air systems as a whole.

The free software package, AIRMaster+, made available by the Department of Energy, allows the modeling of a compressed air system, and the simulation of energy efficiency measures. Using this program, the subject facility's compressed air system was modeled and then energy efficiency measures were simulated. First, an automatic sequencer was simulated to control the system air pressure to a reduced pressure of 95 psig, and to decide which combination of compressors will adequately meet compressed air demand. The second measure was to reduce the air leak load. The reduction of air leaks will reduce the compressor capacity, which then reduces compressor power.

The measures result in cost savings of $52,377 per year, a 128 kW demand reduction, and 804,436 kWh per year in energy reduction. The total project cost is estimated to be $10,800, giving a simple payback of approximately three months.

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A Quantitative Analysis to Determine Methods to Improve an Industrial Compressed Air System

by
Daniel John Paprocki

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APPROVED BY:

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DEDICATION

I would like to dedicate this work to my parents Gerald and Cathi for showing me their support during my extensive stay as a student at North Carolina State University.

BIOGRAPHY

Daniel John Paprocki was born in the year of our lord 1988 in Durham, North Carolina, shortly after Daniel's parents relocated from Milwaukee, Wisconsin. At the age of two, Daniel's family made the decision to move 20 miles from Durham to the burgeoning suburban community of Cary, North Carolina.

During Daniel's elementary school years, he gained an interest in mathematics and sciences, which only grew stronger as he got older. Daniel acquired an appreciation for hands on skills during his formative years. Gerald, Daniel's father would frequently have a home improvement project to work on, and would inevitably recruit Daniel to help out. Additionally, instead of taking the family cars to the shop, Gerald would routinely change the oil and make simple repairs, and always would have Daniel pitch in to acquire these valuable skills. This gave Daniel an appreciation for hard work, the satisfaction of saving money by being self-reliant.

During Daniel's senior year of High School, he applied to North Carolina State University in Raleigh, North Carolina. In the fall of 2007, his first year at N.C. State, he decided to study Engineering and German, and was able to matriculate into Mechanical Engineering after his first year. Mechanical Engineering is a demanding discipline to succeed in, and with a second major in German Studies, Daniel had his work cut out for him. With the double major, graduating in the standard four years would be next to impossible. The German Studies

major required a semester studying in Germany, which would take a whole semester and summer away from engineering. Daniel spent the spring and summer of 2009 in Europe to study in Germany. The semester abroad in Germany proved to be trying, but eye-opening. Immersing in another culture and language is a rare and excellent way of expanding one's horizons, but can be exhausting. While in Germany, Daniel spent time with old friends, met many new and interesting people, saw beautiful cities, artwork, and landscapes.

Once back at N.C. State, Daniel continued to march through his Engineering and German courses. Of the topics within Mechanical Engineering, the thermal sciences interested Daniel the most. During the fall of 2011, Daniel decided to apply for the State Energy Internship Program, which was headed by Dr. Stephen Terry of the Mechanical Engineering Department. This Internship offered Daniel his first hands on engineering experiences. Daniel worked on several measurement and verification projects and was able to attend four energy surveys through the Industrial Assessment Center at N.C. State. After the internship ended, Dr. Terry offered Daniel a position within the IAC.

After graduating in December of 2012 with degrees in both Mechanical Engineering and German Studies, Daniel applied to N.C. State's Mechanical Engineering graduate program, under the direction of Dr. Stephen Terry. Daniel's concentration within Mechanical Engineering is Thermal Sciences. Daniel has attended approximately 40 energy surveys as a member of the IAC, and has been the lead graduate student on several reports.

Outside of school Daniel enjoys lifting weights, running, and going to the gym. Daniel also enjoys collecting and listening to vinyl records. Additionally, Daniel loves spending time with friends and family, playing card and board games, and playing fetch with Seamus.

ACKNOWLEDGMENTS

I would like to thank and recognize several people that helped make this work possible. I would first like to thank my parents, Gerald and Cathi Paprocki for their endless support and patience during my pursuit to earn undergraduate degrees in Mechanical Engineering and German Studies, and a Master's degree in Mechanical Engineering. Undoubtedly, it required a great deal of patience during this length of time. I would also like to thank my brother Nathan, and sisters Katie and Jennifer for their show of support and love during this process. I would like to thank all good friends including, but not limited to, Connor McDonald, Patrick Murray, Sam Gates, Jackson Wooten, Kiran Thirumaran, and Taylor Atkins for their encouragement throughout the process. Additionally, I would like to thank Laura for her encouragement during the writing process.

I would like to state my utmost appreciation to the subject facility for access to their compressed air system, their cooperation throughout the process, and assistance in collecting all necessary data.

I would also like to thank Dr. Stephen Terry, whose support and guidance has not only made this work possible, but has made a Master's degree a realistic possibility for me. I will be forever thankful for the many invaluable opportunities provided to me.

I would also like to thank Dr. Herbert Eckerlin and Alexei Saveliev for serving on my graduate committee advisory committee. I greatly appreciate your participation in this project.

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Chapter 1 – Introduction

Industrial manufacturing, a vital component to the United States economy, requires a significant amount of energy during production. The energy consumed by industry is distributed to a diverse number of end users, which may include ovens, electric boilers, robotics, manufacturing machinery, HVAC, lighting, and, lastly, air compressors. Utilities must provide the energy to manufacturing facilities to run all of these systems, but this comes at a large cost. The substantial energy consumption comes at a price, which cuts directly into profit margins. Compressed air, which is often a necessity to manufacturers, comprises a significant segment of energy consumption in the average manufacturing facility. For the reason that energy costs can directly influence profits and compressed air encompasses a large percentage of the energy use in a manufacturing facility, finding any means to increase efficiency of compressed air systems should be a high priority. This project will analyze an actual compressed air system, and attempt to identify methods to increase the overall efficiency, with a focus on compressor controls, and how to operate a system of compressors efficiently and to meet compressed air demand.

1.1 Electrical Energy and Compressor Use in the United States

The United States relies on several sources of energy for the production of electricity. This includes coal, natural gas, nuclear, hydro, oil, and finally renewables. The percent of generation for each energy source, for 2011, is shown in Figure 1 below (1)

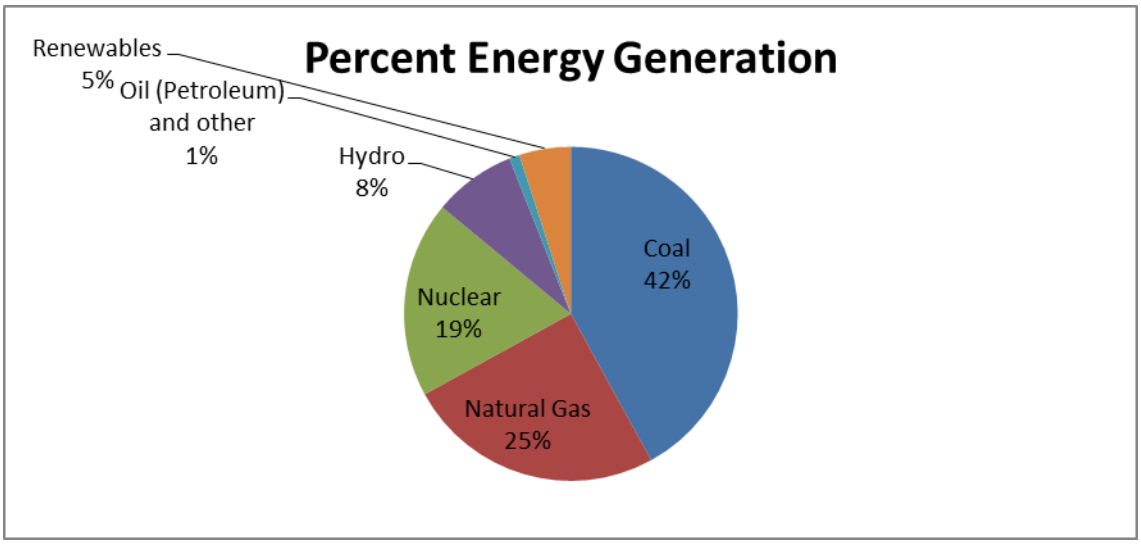


Figure 1: Percent of Energy Generation for Various Sources (1)

As shown above, coal represents the largest percentage of electricity generation, followed by natural gas and nuclear. In 2011, the total U.S. electrical energy consumption was 3,882 billion kWh (1). In addition to electricity, natural gas is also commonly used in manufacturing facilities.

U.S. Manufacturers consume nearly 26% of all electrical energy consumed in the United States (2). Approximately 10% of electricity consumed in manufacturing facilities is from compressed air. Thus, 2.6% of all energy consumed in the United States is consumed by air compressors in an industrial setting (3). This represents a staggering 101 billion kWh per year. To put this into perspective, 101 billion kWh could power the average American home, using 1,200 kWh per month, for over 84 million years. At an average energy price of $0.062/kWh, the total electrical cost to run industrial air compressors is $2.6 billion per year (1). Considering the high energy consumption and energy cost of operating compressed air systems, it would behoove manufacturers to attempt to increase the efficiency of their systems, to capture significant savings.

1.2 Compressed Air Benefits and Drawbacks

It is important to understand why we use compressed air. Compressed air is a fundamental utility at many industrial sites and manufacturing facilities, just as important as power and fuel. Compressed air can have many important uses, including operating pneumatic tools, motors, pneumatic cylinders, automation equipment, conveyors, and controls schemes. There are also many specific compressed air uses in manufacturing processes, including oxidation, fractionation, cryogenics, refrigeration, filtration, dehydration, and aeration (4). A facility may also use compressed air for an application, as opposed to electricity, in a combustible environment, such as a chemical plant. Although compressed air has many appropriate uses, inappropriate uses, in which a more efficient method could be used, can have high costs.

Despite the effectiveness and flexibility of compressed air, unfortunately the overall efficiency of a typical compressed air system is only 10% to 15%. This is due to losses from the heat of compression, meaning that approximately 80% of the electrical energy consumed by air compressors is converted to heat and not compressed air. Thus, the use of electricity instead of compressed air is much more efficient. To illustrate this issue, we will compare the operating cost of a one hp compressed air motor to a standard one hp electric motor. A typical one hp compressed air motor requires 30 scfm at 90 psig, which requires approximately 7 hp at the compressor shaft. Therefore, the compressed air motor will require 7 times as much electrical input, and money, to produce the same amount of work as a standard one hp motor. This indicates that one should be judicious when determining whether or not compressed air should be used for a certain task at a manufacturing facility.

1.2.1 Inappropriate Compressed Air Use

Considering the expensive and inefficient nature of compressed air as a utility, inappropriate compressed air uses must be kept to a minimum. An inappropriate compressed air use is defined as any application that can be done more efficiently by a method other than compressed air (5). Provided below is a table from the Industrial Technologies Program, which lists potentially inappropriate uses and a suggested alternative to that use.

Table 1: Inappropriate Uses of Compressed Air and Alternative Methods (5)

Potentially Inappropriate Uses	Suggested Alternatives/Actions
Clean-up, Drying, Process Cooling	Low-pressure blowers, electric fans, brooms, nozzles
Sparging	Low-pressure blowers and mixers
Aspirating, Atomizing	Low-pressure blowers
Padding	Low to medium -pressure blowers
Vacuum generator	Dedicated vacuum pump or central vacuum system
Personnel cooling	Electric fans
Open-tube, compressed air-operated vortex coolers without thermostats	Air-to air heat exchanger or air conditioner, add thermostats to vortex cooler
Air motor-driven mixer	Electric motor-driven mixer
Air-operated diaphragm pumps	Proper regulator and speed control; electric pump
Idle equipment	Put an air-stop valve at the compressed air inlet
Abandoned equipment	Disconnect air supply to equipment

1.3 Introduction to Compressor Controls

Compressor controls can vary from compressor to compressor, and can be unique to a compressor system based on the number and types of compressors the system is comprised of. As the number of compressors in a system increases, so does the complexity of the required controls.

Single air compressor systems can consist of two distinct compressor types, those being positive displacement and dynamic compressors. Typically, positive displacement machines are controlled by on/off, load/unload, modulation, or VFD control types.

On/off controls are generally found in smaller reciprocating compressors. When the desired system pressure is reached, the reciprocating compressor simply shuts down. The compressor will subsequently turn back on when the system pressure reaches a set minimum allowable pressure. For larger rotary screw compressors, load/unload and modulation are commonly employed. Using load/unload controls will track system demand and help save energy while unloaded. This will also ensure that the compressor does not turn on and off in short cycles, which can destroy larger motors from locked-rotor current. Load/unload controls allow the compressor to unload when the system pressure reaches a predetermined maximum.

Modulation controls are typically found in rotary screw and dynamic compressors. Modulation follows system demand by restricting the flow of air to the compressor through the use of an inlet valve, such as a butterfly valve. As less air flows through the inlet, less power is required to compress that air. However, the main drawback of using modulation is that it reduces the pressure of the inlet air, causing the compression ratio to increase. For flooded oil rotary screw compressors utilizing modulation, the percent kW input at 50 percent capacity will likely be approximately 85%, which is rather inefficient. Allowing less air through the inlet increases the compressor efficiency, but the decreased pressure at the inlet is akin to taking a step backward.

Variable frequency drives are generally found in rotary screw compressors. The variable frequency drives allow the motor to track system demand by altering the speed of the electric

motor by varying voltage frequency. The motor speed and percent power have approximately a one to one ratio, meaning that at half the fully rated revolutions per minute the motor will draw half the fully rated power input. Compressor capacity and motor speed also have a one to one ratio, which implies that at half capacity the compressor will only require half of its fully rated power. This is a rather efficient method of controlling a single compressor.

As compressor systems grow larger, it is important for compressors to communicate with each other, helping to ensure compressors only turn on when necessary. Generally network controls are utilized to make certain compressors communicate with each other. More complicated systems, such as those with multiple compressor rooms consisting of both centrifugal and positive displacement compressors, require System master controls. System master controls allow for the control of large compressed air systems through measurement of system parameters; pressure for instance. Further description of each of the aforementioned control methods is detailed in Chapter 3.

1.4 Main Project Objective

The main objective of this study is to collect data from a subject facility's compressed air system using data loggers, analyze data, and then to model the compressed air system using AIRMaster+, which is a free software package made available by the Department of Energy. After the compressed air system is modeled in AIRMaster+, the software will simulate various energy efficiency measures, one of which is using an automatic sequencer. An

automatic sequencer is essentially system master controls, which help operate lager compressed air systems efficiently. To understand how multiple compressors operate together, first a familiarity of different types of compressors and an understanding of various methods of single compressor and multiple compressor controls must be realized.

Through the simulation of the implementation of an automatic sequencer using AIRMaster+, the potential energy and cost savings associated with sophisticated compressed air system controls can be conveyed to the subject facility.

Chapter 2 –Compressed Air

Modern manufacturing facilities utilize multiple types of air compressors to meet the compressed air demand for their processes. Currently, the most common compressors used in the industrial world are reciprocating compressors, rotary screw compressors, and lastly, centrifugal compressors. Based on the application and compressed air demand of the manufacturing facility, a type of compressor is chosen. Each compressor has its own benefits and drawbacks, which must be considered before purchase and installation. This chapter will discuss compressed air terminology, common compressed air components, and types of compressors.

2.1 Important Compressed Air Terminology

In order to gain a better understanding of compressed air systems, it is important to become familiar with terminology related to compressed air.

Capacity: Capacity is the amount of air delivered under specific conditions. This is usually measured in cubic feet per minute, or CFM (6).

Cubic Feet Per Minute (CFM): This is the volumetric flow rate (6).

> **Actual CFM (ACFM):** Flow rate of air at a certain point at the actual temperature and pressure at that point. When this is used for the capacity of an air compressor, it is measured at prevailing ambient conditions of temperature, pressure, and relative humidity (6).

Inlet CFM (ICFM): The volumetric air flow rate through the compressor inlet valve under the prevailing ambient conditions. For positive displacement machines ICFM and ACFM should be identical, but could be different in some centrifugal air compressor designs due to air losses through shaft seals (6).

Standard CFM (SCFM): The volumetric flow of free air measured and converted to a standard set of reference conditions. The International Standards Organization (ISO) defines standard air as 14.5 psia, 68°F, and 0%relative humidity (6). This is equivalent to specifying mass flow rate, since a volume at a given temperature and pressure has a specific density.

Demand: The CFM of air required by a specific point in a facility, or by the entire facility. This is generally referenced to scfm.

Humidity, Relative: Relative humidity is the ratio of the actual vapor pressure to the vapor pressure if the air were completely saturated (7).

Dew Point: The dew point is the temperature at which water vapor will begin to condense out of air if the air is cooled at constant pressure.

Specific Humidity: The mass of water vapor in an air vapor mixture per mass of dry air.

Power: Power is work over a period of time. Power is often measured in measured in kW, or brake horsepower.

Brake Horsepower (bhp): This is the horsepower required at the compressor shaft to produce compressed air.

Load Factor: Load factor is the average compressor load divided by the maximum rated compressor load over a period of time.

Full-Load: When the air compressor is operating at full speed with a fully open inlet and delivering maximum air flow.

Specific Power: A method of measuring compressor operating efficiency, usually in the form bhp/100 ACFM, or kW/100 ACFM.

Total package Input Power: This is the total power used by the air compressor, including the drive motor, fans, motors, and controls.

Pressure: Pressure is defined as force per unit area. This is commonly measured in pounds per square inch (psi).

Atmospheric Pressure: This is the naturally occurring pressure in the atmosphere. The atmospheric pressure at sea level is approximately 14.7 psi.

Gauge Pressure: Pressure determined by instruments, which are calibrated so that atmospheric pressure is zero psi. Gauge pressure is expressed as psig.

Pressure Drop: Pressure drops occur in compressed air systems due to friction or restrictions.

Pressure Range: The range between minimum and maximum pressures for an air compressor. Also referred to as load-no load pressure ranges.

Rated Pressure: The ideal pressure for optimal compressor performance.

Receiver: A pressure vessel used to store compressed gas or air.

Surge: A dangerous and destructive operating condition for centrifugal air compressors. This occurs when a reduced flow rate results in backwards flow. The compressor can no longer overcome backpressure.

2.2 Types of Compressors

Industrial compressors are divided into two main types of compressors, which are positive displacement and dynamic. For the positive displacement compressor, a finite quantity of air or gas enters into a compression chamber and the volume of the chamber is mechanically reduced, thereby increasing the pressure of that gas before discharge (4). Dynamic compressors, such as centrifugal compressors and axial flow compressors, operate much differently. By means of impellers rotating at an extraordinary speed, a dynamic compressor imparts kinetic energy to continuously flowing air. The kinetic energy of the air or gas is changed into potential energy (pressure) by the impellers and diffusers (4). Below is a figure that further breaks air compressors into subcategories.

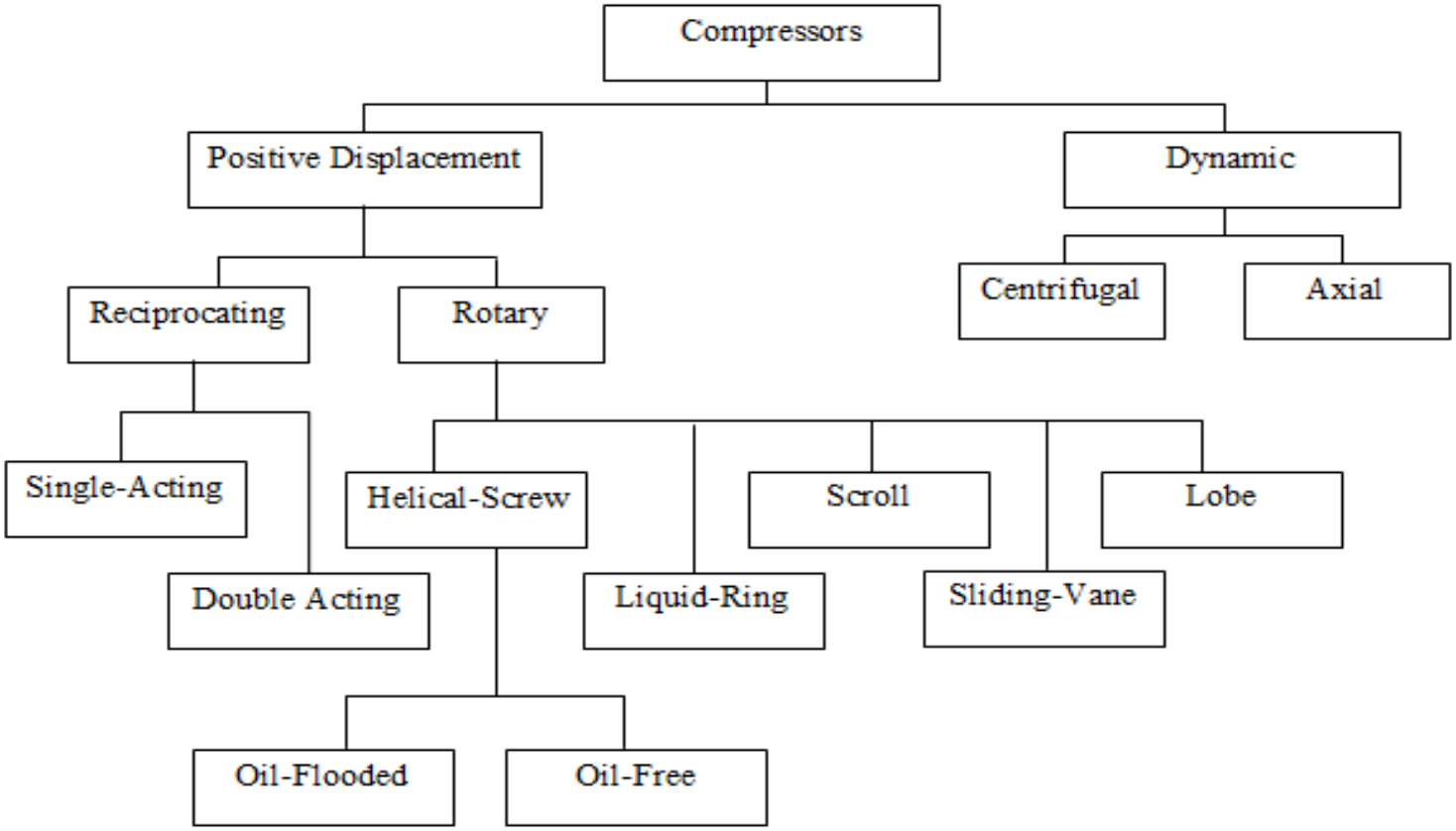


Figure 2: Compressor Subsets (2)

2.3 Positive Displacement Compressors

Positive displacement compressors are available in two distinct categories, which are reciprocating and rotary screw compressors. Reciprocating compressors are divided further into single-acting and double-acting, both of which operate similar to that of a bicycle pump to compress air. Industrial facilities also commonly use rotary screw compressors, which compress air by trapping air inside the rotors and compress the air as it travels down rotors to the discharge point (4). Rotary screw compressors are often oil flooded to lubricate the rotors, but oil free is also available.

2.3.1 The Reciprocating Compressor

Typically, the modern reciprocating compressor used in manufacturing facilities are between 5 hp and 30 hp. Single acting reciprocating compressors are generally available up to 150 hp and can produce higher than 175 psig compressed air. (8) For a single acting reciprocating air compressor, the operating efficiency is between 22 and 24 kw/100 CFM. In general, a double acting reciprocating compressor can achieve an operating efficiency of 15 to 16 kW/100 CFM.

Reciprocating compressors are often staged to improve efficiency, with an intercooler between stages. Most reciprocation compressor systems have two stages to produce 100 psig

air. Three or more stages may be used where high pressure (greater than 150 psig) is needed, such as in blow molding operations.

Figure 3 below is a cross section of a reciprocating compressor with three pistons to produce compressed air.

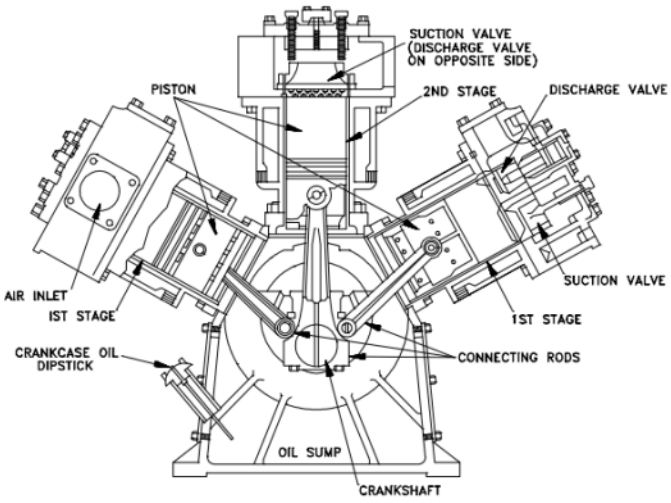


Figure 3: Reciprocating Compressor Cross Section (9)

The single acting reciprocating compressor is distinguished by a piston and cylinder, similar to that of an internal combustion engine, which is driven by a connecting rod from the crank (4). The reciprocating compressor is essentially a piston cylinder device with an inlet and exit valve. The compression cycle starts when the piston is at top dead center, when the piston volume is zero, not including the clearance volume.

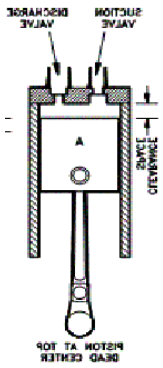


Figure 4: Piston at Top Dead Center

As the crank shaft turns, the piston moves down in the cylinder, thereby increasing the piston volume and creating a vacuum. The intake valve allows atmospheric air to enter the chamber during this process (3).

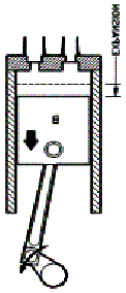


Figure 5: Air Intake

At the intake valve, atmospheric pressure is higher than the pressure in the cylinder, therefore air enters the cylinder. At bottom dead center, the intake valve is closed, and the piston is driven back up the cylinder by the crank shaft (3).

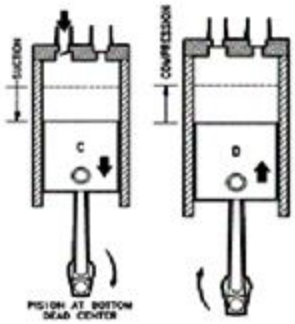


Figure 6: Piston at Bottom Dead Center

The volume in the cylinder decreases as the piston moves towards top dead center, which increases the pressure. At the desired gauge pressure, the exhaust valve opens and the compressed air is released from the cylinder. The desired compressor pressure is often controlled by a spring, which will force the exhaust valve shut. The spring may be adjusted to allow for different pressure settings (3).

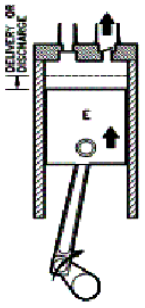


Figure 7: Top Dead Center

At the end of the cycle the compressed air is released, and both the intake and exhaust valves are closed. The cycle repeats until the demand for compressed air is satisfied, at which point the compressor will shut off. Commonly the power to drive this cycle is derived from an electric motor. Figure 8 below details the complete compression cycle (3).

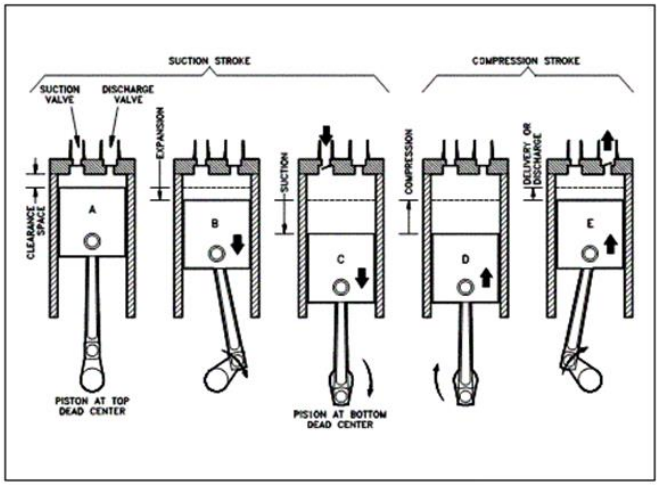


Figure 8: Compression Cycle

The double acting reciprocating compressor is similar to the single acting reciprocating compressor, but with one exception. Double acting means that the compressor uses both sides of the piston and cylinder for air compression, effectively doubling the capacity for a giving cylinder size. This type of compression is particularly efficient with multi-stage compressors (4).

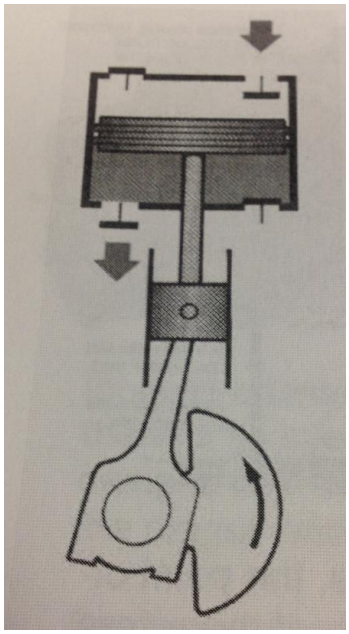


Figure 9: Double Acting Reciprocating Compressor (4)

Using a reciprocating compressor to produce compressed air can have advantages. Generally, reciprocating air compressors are small in size and weight, and therefore can be located close to the point of use. This would avoid long lengths of compressed air piping and potential pressure drops. Also, reciprocating compressors generally require simple maintenance procedures. Unfortunately, reciprocating compressors are associated with a high initial cost, and high vibrations, which require a thick foundation (4).

2.3.2 Rotary Screw Compressors

Oil flooded rotary screw compressors and oil-free rotary screw compressors are two common types of rotary screw compressors. The more common air compressors found in industry

today is the oil flooded rotary screw compressor, due in part to its versatility (4). The operation of the screw compressor is distinctly different than the aforementioned reciprocating compressor. The rotary screw compressor mechanically compresses air with two screws, one of which, the male screw, is driving the female screw. These screws are meshed together in a stator and rotate. Air flows through the inlet port and becomes trapped between the meshing screws. As the screws rotate, the point of intermeshing, where the air is trapped, moves gradually along the axial length of the rotors. As this occurs the space occupied by air reduces in volume, resulting in an increase in pressure. Air compression follows until the air reaches the discharge port and the air is released to the demand side of the compressed air system (4). Figure 10 depicts the two screws meshed together.

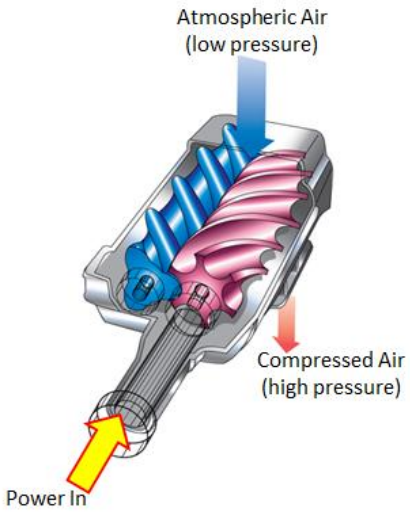


Figure 10: Rotary Screw Compressor (3)

Lubrication is vital to health and longevity of the oil-flooded rotary screw compressor. The lubrication serves three basic functions. The oil lubricates the meshing rotors and bearings, and serves to intercool the air during compression. The lubrication also performs much like oil in an automobile piston and cylinder system would, in that it acts as a clearance between the meshing rotors. Thus, the rotors never touch, greatly reducing friction and heat (4). Commonly, oil flooded rotary screw compressors are available from 3 hp to 900 hp, with discharge pressures from 50 psig to 250 psig (4). Figure 11 is a schematic of a generic oil-flooded screw compressor package.

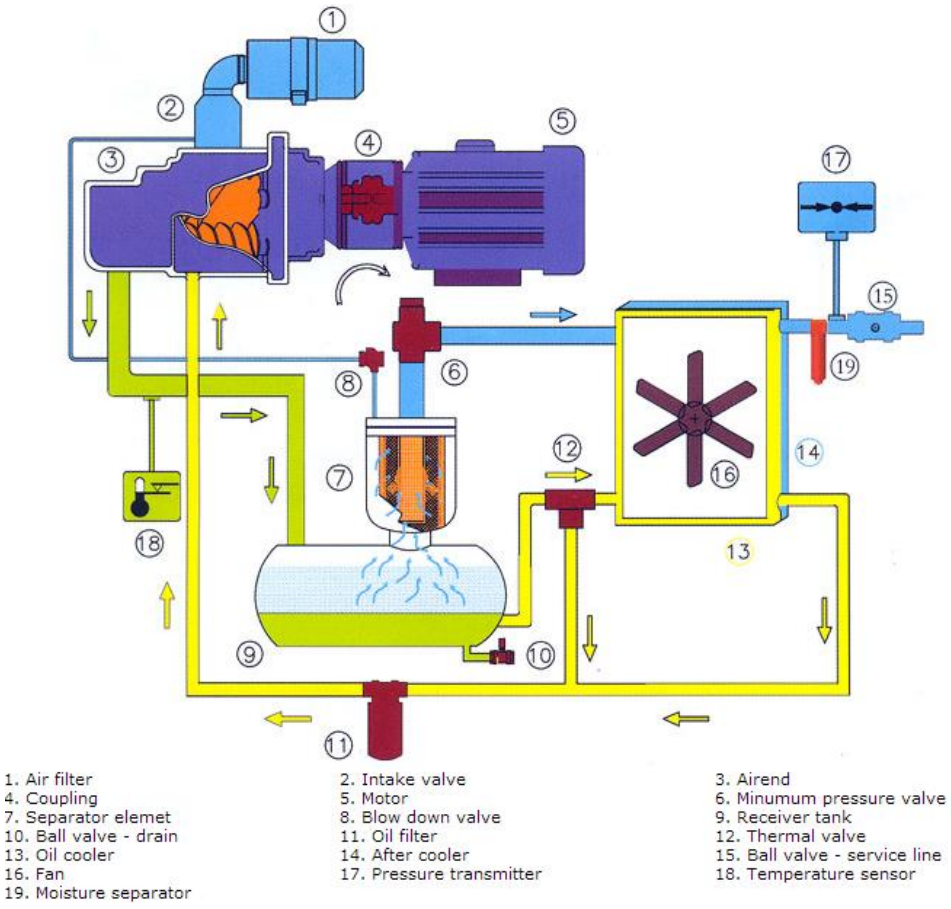


Figure 11: Oil-flooded Screw Compressor (10)

Advantages of oil-flooded rotary screw compressors include relatively compact sizes for high horsepower systems, low vibration, and accurate part load capacity control systems. Disadvantages of an oil-flooded rotary screw compressor include the fact that lubricant can carry over into the compressed air flow, and the system efficiency can vary depending on the chosen control mode. One can expect to achieve operating efficiencies of 17 to 22 kW/100 CFM for single stage compressors, and 16 to 19 kW/100 CFM for two stage compressors (4).

The lubricant-free rotary screw compressor works in the same fashion as the oil-flooded screw compressors. As the name would suggest, there is no lubrication injected in the compression chamber. Additionally, there are two distinct types of oil-free rotary screw compressors; dry type and water injected type. Figure 12 illustrates an oil-free rotary screw system, with the distinct lubricated timing gears.

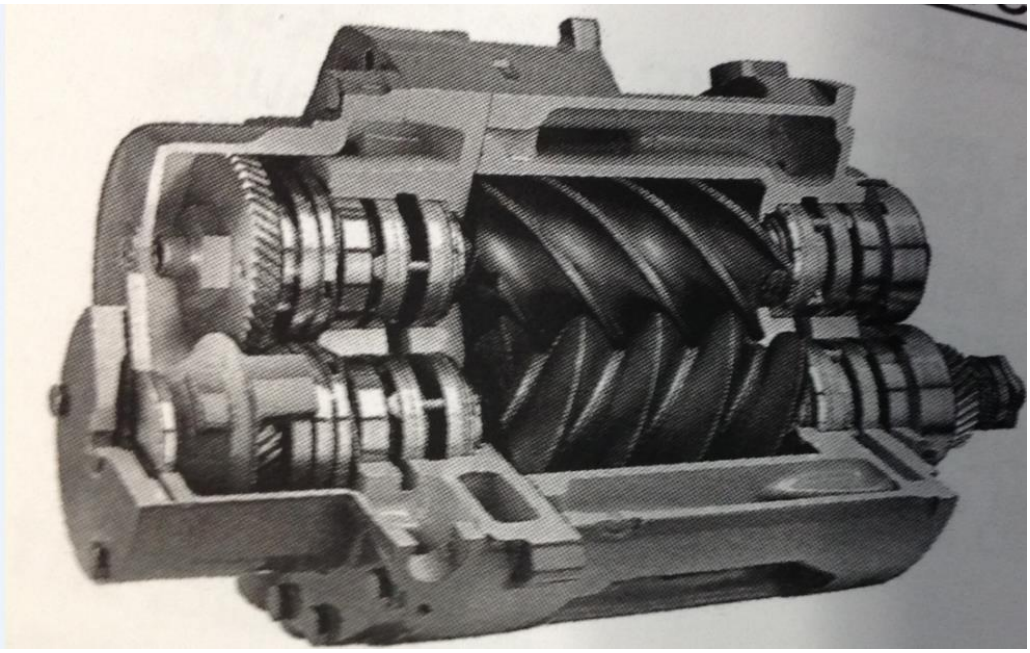


Figure 12: Oil-free (4)

A dry type oil-free screw compressor uses lubricated timing gears, which are external to the compression chamber, to keep the intermeshing rotors from touching. These types of compressors do not have coolant injected into the compression chamber, and therefore may require two stages of compression, with an intercooler between stages and an after cooler

after the second stage, to compress air to higher pressures. This is similar to a reciprocating compressor. Teflon may be used to help seal the rotors and limit friction between moving parts. A one stage dry type compressor can operate up to 50 psig, whereas a two stage may operate up to 150 psig.

Similarly, the water type lubricant-free rotary screw compressor uses timing gears. However, in this type of compressor water is injected into the compression chamber. This acts to remove the heat of compression and seal any internal clearances. An oil-free screw compressor can be expected to operate at an efficiency of 18 to 22 kW/100 CFM. Although these types of compressors produce oil-free compressed air, they have a higher initial cost, are generally less efficient and require higher maintenance costs than their oil-flooded counterparts (4).

2.4 Dynamic Compressors

The most common and widely used compressor used for large industrial applications is the centrifugal compressor. Centrifugal compressors operate by converting the high velocity of air flowing through an impeller to pressure energy. The impeller accelerates the continuously flowing air stream to a high velocity, and then the compressor converts the kinetic energy to pressure energy as the speed is reduced by means of a diffuser (4). Interestingly, as the system pressure decreases, the compressor capacity to produce compressed air increases (4). A centrifugal compressor will operate at an efficiency of 16-20 kW/100 CFM. (4) Also, it

should be noted that a centrifugal compressor will produce oil-free compressed air. Figure 13 depicts an impeller accelerating the flow of air through a compressor.

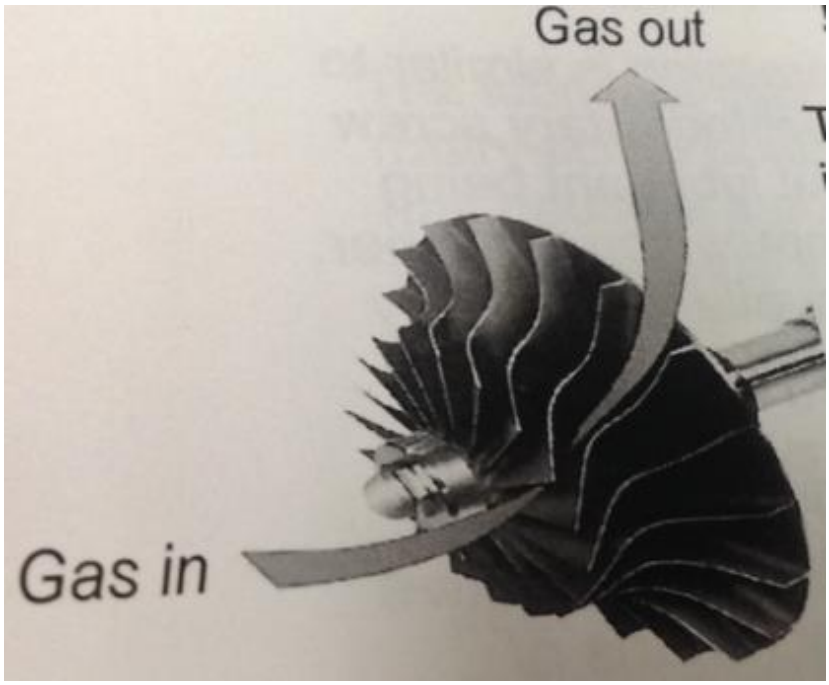


Figure 13: Impeller of Centrifugal Compressor (4)

Chapter 3 – Compressor Controls

The purpose of compressor controls is to match the compressor output with the facility compressed air demand. This is done by sustaining the compressor discharge pressure between a specified range. Developing a control strategy, whether for one compressor or multiple air compressor systems, is vital to saving energy and money. First, controls for individual compressors will be discussed, followed by multiple compressor system controls.

3.1 Basic Individual Compressor Controls

For smaller single compressor compressed air systems, controls are contained to the compressor itself. Individual compressor types to be discussed are start/stop, load/unload, modulating, dual/auto, variable displacement, and variable frequency drive control.

3.1.1 Start/Stop Control

For reciprocating compressors and rotary screw compressors under 25 hp, a simple Start/Stop control scheme would be a satisfactory control method. The compressor motor turns off as a specified pressure set point is reached and then turns back on when the pressure drops below a given lower pressure set point.

A simple example of start/stop control is a home thermostat. During the winter, as the temperature in a space dips below a set point temperature, the heating system will turn on to supply heat. The temperature in the space will rise until it reaches another set point, at which point the heating system will shut off. In this manner an average temperature is maintained. The difference between the cut on point and cut off point is the deadband. The system will operate between the two setpoints in this deadband region.

Depending on storage capacity, the pressure range, or deadband, needed for this control method can be as high as 35 psi. This is a fairly simple control scheme, needing only a pressure switch. Furthermore, this method can save energy, as the motor and compressor operate only when required. However, the frequent full load amp starting can wear down a motor, and can only be used with smaller motors.

3.1.2 Load/Unload Control

Load/Unload controls are a common control scheme for oil flooded rotary screw compressors, but can be used for larger reciprocating and centrifugal compressors. As the predetermined pressure set point is reached, the compressor is allowed to unload, which uses a lower power setting and saves energy. To unload means to close the inlet air damper, ceasing the production of compressed air, and slowly depressurizing the compressor. During this process, the compressor is still pushing against the pressure in the sump, which requires power, but the sump pressure is allowed to slowly decrease. Once the sump reaches about 15

psig, the compressor operates at fully unloaded conditions and draws 30% to 40% of its full load power.

Decreasing the oil pressure too quickly would be analogous to shaking up an unopened soda bottle. Shaking the soda bottle causes dissolved carbon dioxide in the liquid to be released thus, increasing the pressure. If opened too quickly, the soda will foam and result in a mess. The same phenomenon will occur in an oil flooded rotary screw compressor. Oil is compressed along with air, and gas is dissolved in the lubricant. If the pressure is reduced too quickly, the oil will foam and lose its ability to lubricate the rotors. For this reason the compressor is blown down slowly. Figure 14 depicts the load/unload cycle of a real 100 hp compressor.

In Figure 14 below, the compressor is fully loaded until approximately 13:38:10, at which point the compressor unloads because the maximum system pressure set point is reached. From 13:38:10 to 13:39:40 the compressor blows down and gradually uses less power. At approximately 13:39:40 the compressor reloads when the system pressure falls to the minimum pressure set point. At this point, the compressor demand rapidly increases until it has reached full load. The compressor remains fully loaded until the desired system pressure has been reached.

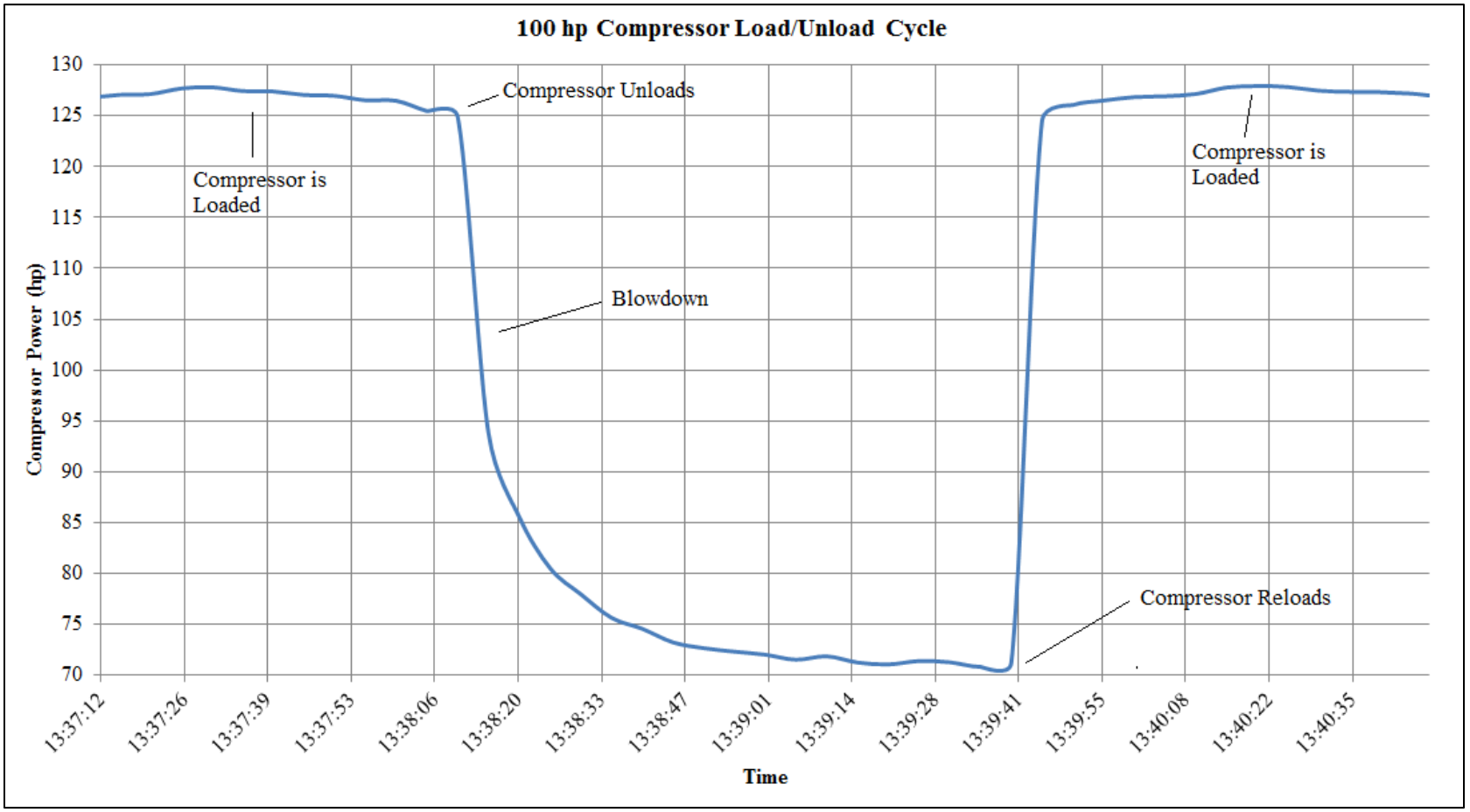


Figure 14: Load/Unload Cycle

Thus, some period of time is required to fully unload a rotary screw compressor. As the pressure in the system drops to a fixed lower pressure, the compressor reloads. This method of controlling higher powered compressors is advantageous compared to start/stop controls because there is less stress on the electric motor, increasing the longevity of the compressor package. However, if the compressed air system does not have enough storage, short cycles may occur. Short cycles may cause premature wearing and energy savings may be reduced.

Compressed air storage is key to running an efficient load/unload control scheme. The compressor will be more efficient with increased storage. For example, if a compressed air system has one gallon/CFM of storage and is operating at 50% capacity, the compressor will still be using approximately 84% of its kW input on average. This operating condition, the result of low air storage, arises because the compressed air is depleted at a quick rate and the compressor will need to load before the sump fully depressurizes and it is fully unloaded. The compressor will begin to unload and decrease compressor power. However, before the compressor can completely unload at 30% to 40% of its power, the compressor will reload due to the fast depletion of compressed air in the system. Consequently, the compressor will operate at a higher average percentage of its full load power during operation.

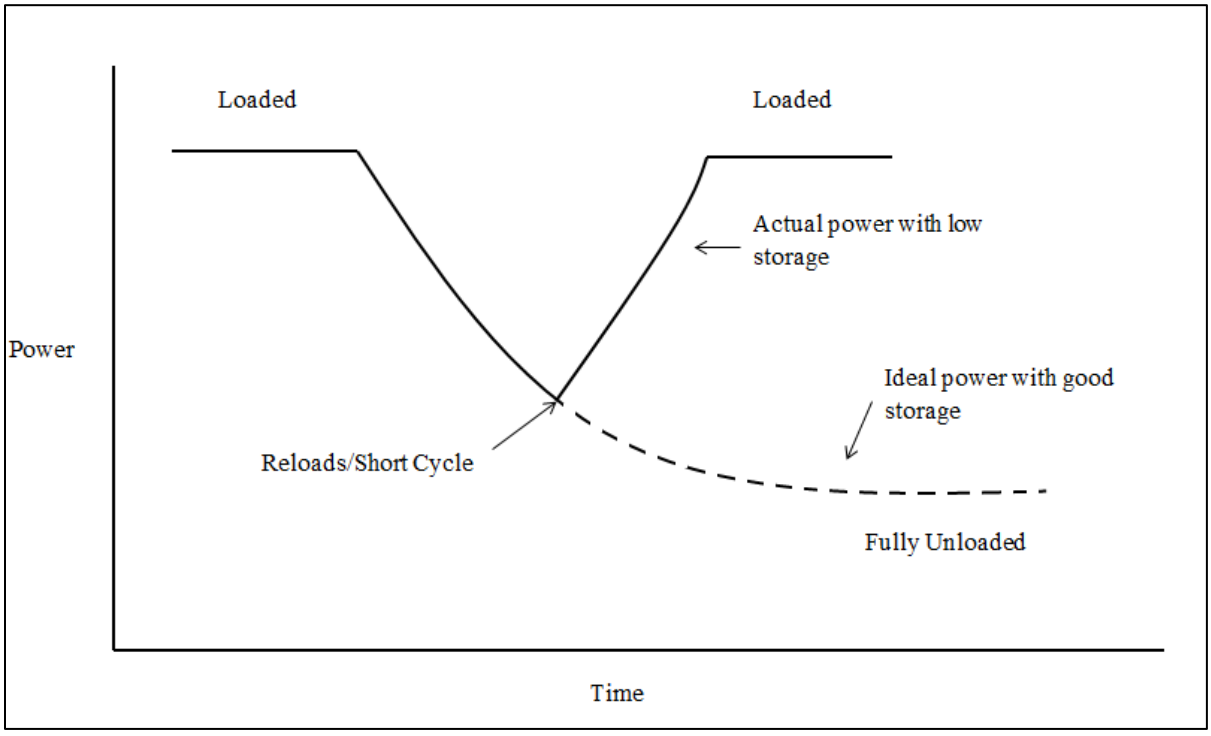


Figure 15: Short Cycle

Conversely, should a rotary screw compressor, using load/unload controls, have 5 gallons/CFM, the compressor would be consuming approximately 68% of its kW input. With more storage, the compressed air in the system will deplete at a slower rate and the compressor will have more time to remain unloaded, and therefore will use less energy. Storage should be sized to ensure additional compressors will not need to be turned on in a large demand event. Figure 16 plots the efficiency of running a load/unload control scheme for a rotary screw compressor at 1 gallon of storage per CFM to 10 gallons of storage/CFM (4) (11).

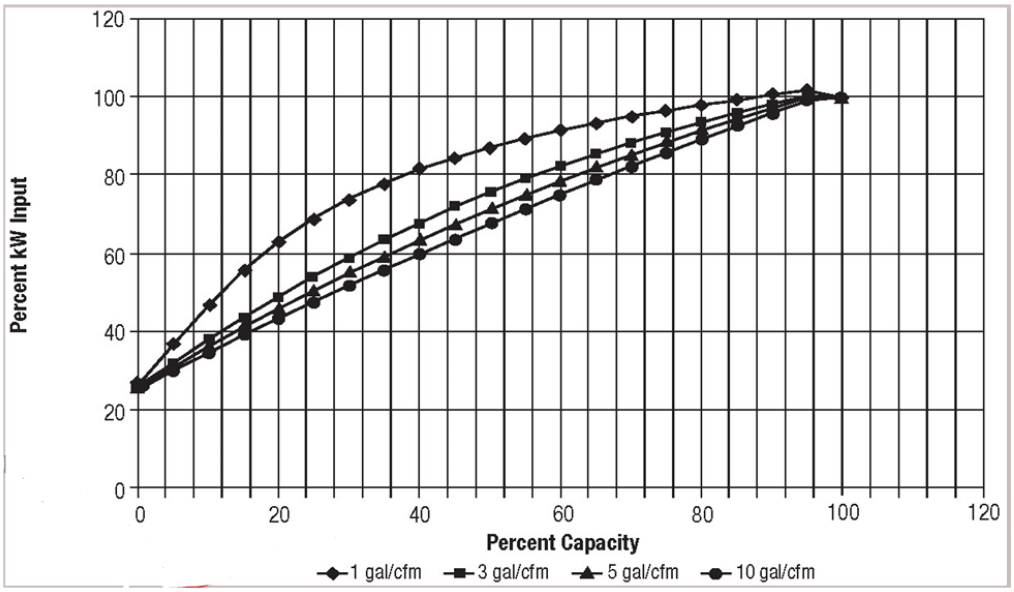


Figure 16: Average kW vs Average Capacity with Load/Unload Capacity Controls (11)

3.1.3 Modulating Control

Modulating controls are often used in industry for rotary screw and centrifugal compressors. Modulation restricts the air flow through the inlet valve of the compressor to reduce the production of compressed air. This allows for tighter pressure control and continuous motor operation, reducing wear. Modulating also allows for accurate matching of capacity production to compressed air demand. The main issue with modulation is that the pressure ratios increase as the inlet valve is restricted, causing inefficient operation. As the inlet valve closes the pressure at the inlet decreases accordingly. For example, a compressor using modulating controls with an inlet valve 40 percent open will experience an inlet air pressure of 40 percent of standard atmospheric pressure. Atmospheric pressure at sea level is approximately 14.7 psia, therefore the inlet air pressure for the aforementioned operating condition will be 5.88 psia. If the compressor is producing compressed air at 100 psig, the new compression ratio is 17.5:1 as opposed to 7.8:1 (11). This is illustrated in Figure 17.

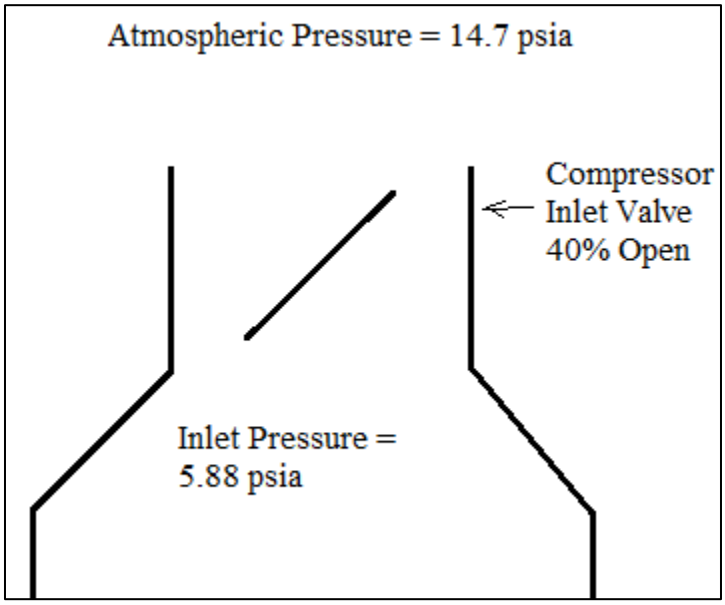


Figure 17: Compressor Inlet Butterfly Valve 40% Open

The result is that a modulating rotary screw compressor will require approximately 88% of the total kW input to produce 50% of its capacity (4) (11). The energy savings come from the reduced mass of the air being compressed. The following curve in Figure 18 illustrates the relationship between percent kW input power and percent capacity for a rotary screw compressor utilizing modulating controls.

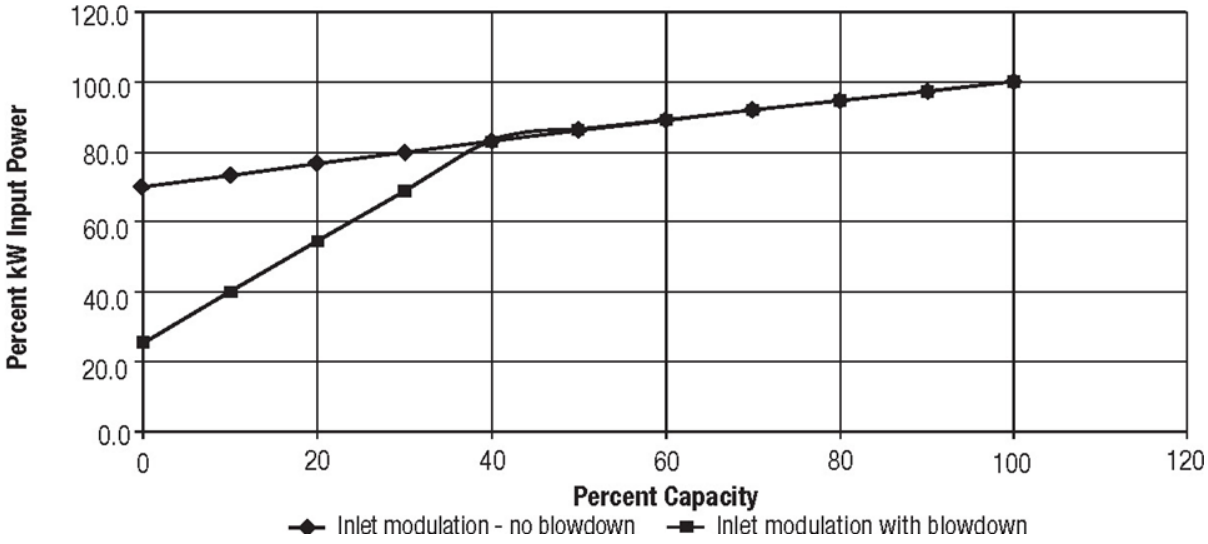


Figure 18: Percent kW Input Power vs. Compressor Capacity for Modulation (11)

Modulating controls might be appropriate if a rotary screw compressor is operating with little to no storage at higher capacity. Generally, rotary screw compressors can be switched to run on either load/unload or modulating controls. Some compressors may allow modulation with blowdown, which is a control method that allows the compressor to unload at some low capacity. From Figure 18 above, a compressor using inlet modulation with blowdown would use modulating control until about 40 percent capacity, at which point the compressor begins using load/unload control.

3.1.4 Dual/Auto and Variable Displacement

The next control type is the Dual/Auto, which is for either small reciprocating compressors or rotary screw compressors. For reciprocating compressors, dual/auto dual allows the compressor to select either start/stop or load/unload. Dual/Auto Dual controls allow oil-flooded rotary screw compressors to select between modulating and load/unload controls. Furthermore, if unloaded for a long duration, this control type will shut down rotary screw compressors.

Another control type available for rotary screw compressors is variable displacement. Variable displacement effectively shortens the length of the screws by using a turn-valve, spiral-valve, or a poppet-valve. This allows a decrease in the amount of air flowing through the inlet, and in turn decreases the amount of power needed to compress air. This is an effective way to increase and decrease compressor capacity. Variable displacement is generally more efficient at running partially loaded compressors than even load/unload controls with high storage. Figure 19, below, illustrates the relationship between percent kW input power and percent capacity for a rotary screw compressor with variable displacement control (4) (11).

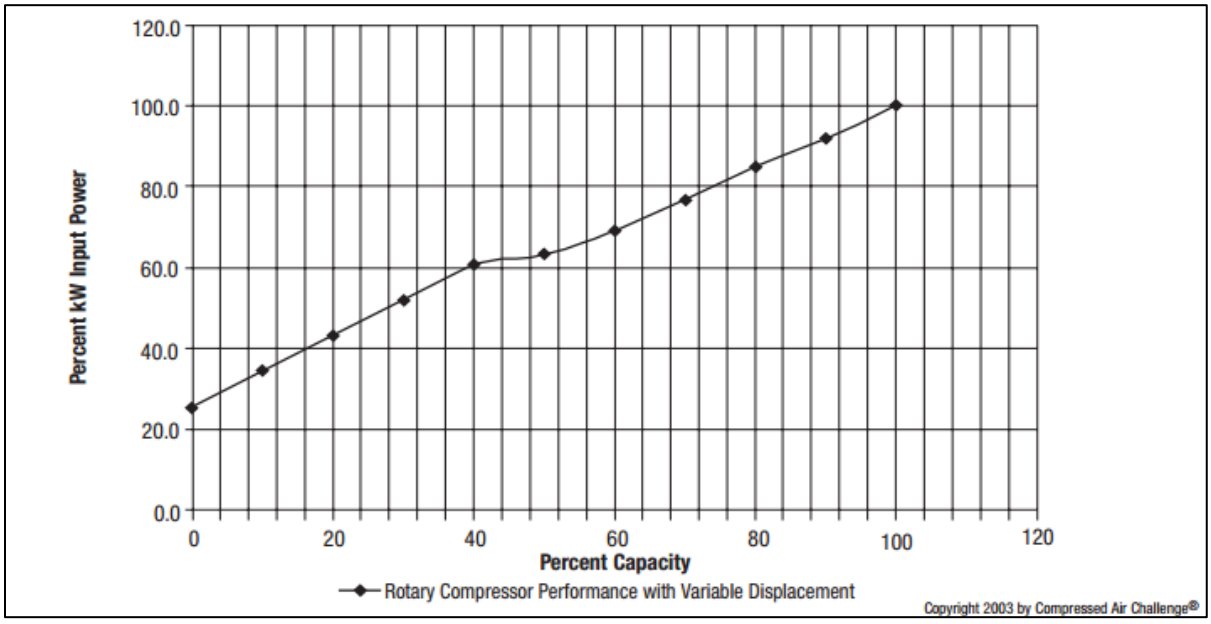


Figure 19: Variable Displacement Control (8)

3.1.5 Variable Speed Drive Control

The most effective control method for operating at partial loads is using a variable speed drive (VSD). The variable frequency drive adjusts the compressor capacity by changing the speed of the electric motor as compressed air demand in the system changes. The compressor capacity is proportional to the speed of the male rotor, but due to the design of variable displacement drive package, at full load capacities the male rotor is rotating above the optimum rotor speed. Thus, a compressor with a VSD will require more power at full load than otherwise, but a VSD offers significant power reduction and energy savings at lower loads. Figure 20 illustrates the relationship between percent kW input power and percent capacity for an oil-flooded screw compressor with a variable speed drive (11).

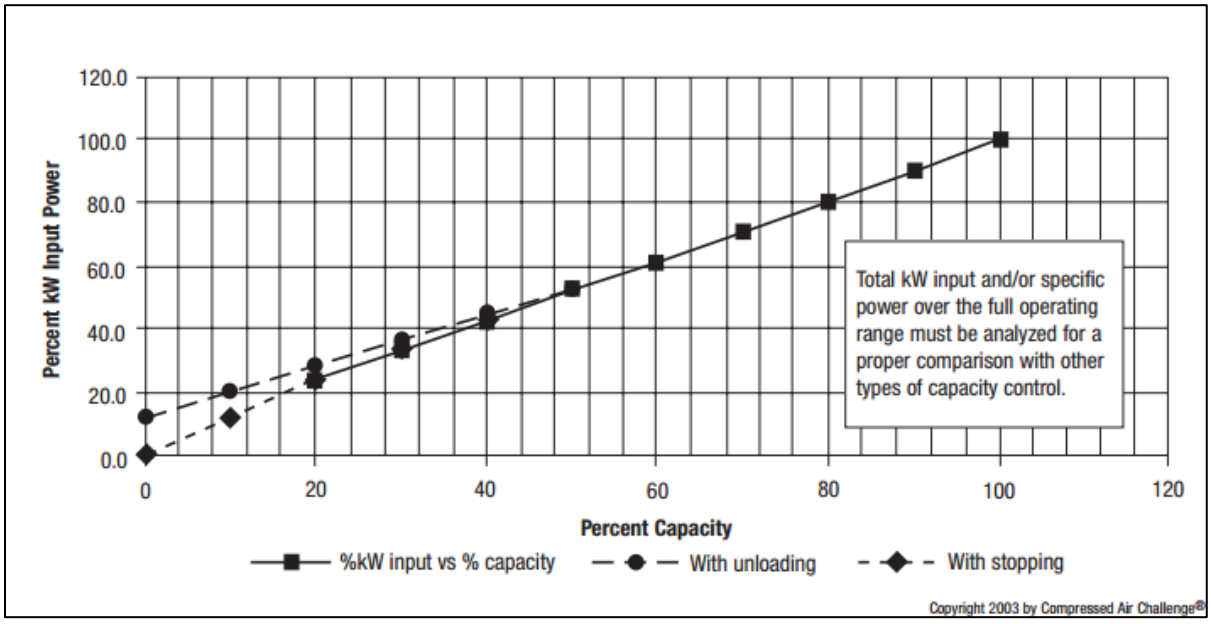


Figure 20: VSD Curve (8)

3.2 Centrifugal Compressor Operation and Control

The operation characteristics of centrifugal air compressors are complex and affected by inlet air density and intercooler cooling water temperature. The basic compressor performance curve, pressure against flow, is determined by the design of the impeller. An example of this is that an impeller with radial blades will yield a low rise in pressure as flow is decreased, and backward leaning blades will create a higher rise in pressure as flow is decreased.

When operating centrifugal compressors it is important to control for surge and choke. Surge, which is harmful to the machine, occurs when flow reverses in the diffuser after the air leaves the impeller. This is possible because of an increased flow path length in the diffuser,

causing the flow to dissipate due to friction and ultimately the flow reverses (11). The aerodynamic instability within the system is to the extent that the compressor can no longer deliver the necessary pressure to produce flow downstream (12). To avoid the destructive surge condition, centrifugal compressors may use discharge bypass or blow-off control. To avoid surge, enough compressed air is discharged to atmosphere to keep the unit at some minimum load, while the required capacity is delivered to the facility. For example, if the facility needs only 45% capacity, the compressor will produce approximately 70% of its capacity and blow-off the extra 25% of the compressed air. For this reason, blow-off is quite wasteful and expensive, and therefore should be avoided. Figure 21 depicts the surge line in a centrifugal compressor performance curve.

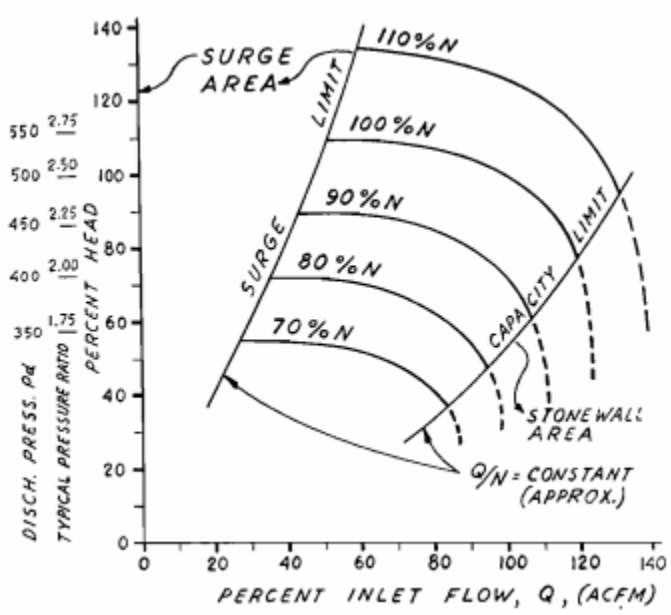


Figure 21: Centrifugal Compressor Performance Curve (12)

The opposite of surge is choke, or stonewall. This occurs at flow rates that are above the design rate, which should not occur until the velocity at the impeller inlet reaches the speed of sound. As the compressor exceeds the capacity limit, the performance enters the choke area. At this point, any increase in flow rapidly decreases the pressure being produced (11).

Centrifugal compressors are designed to operate at pre-determined tip speed, which is usually between Mach 0.85 and Mach 0.9. Thus, to increase and decrease the flow rate, an inlet throttle valve is utilized. Additionally, the throttle valve reduces the pressure, and air density at the inlet before the impeller, which reduces the head produced by the impeller. Throttle valves can usually control capacity of centrifugal compressors from 100% to about 70% of full capacity. Properties such as air density can affect the capacity of the air compressor. For example, higher density cool air will effectively increase the volumetric flow rate at any compressor capacity. Though the capacity can be increased with cooler inlet temperatures, this also results in an increase in power consumption (11). In Figure 22, the effect of inlet air temperature on capacity is illustrated.

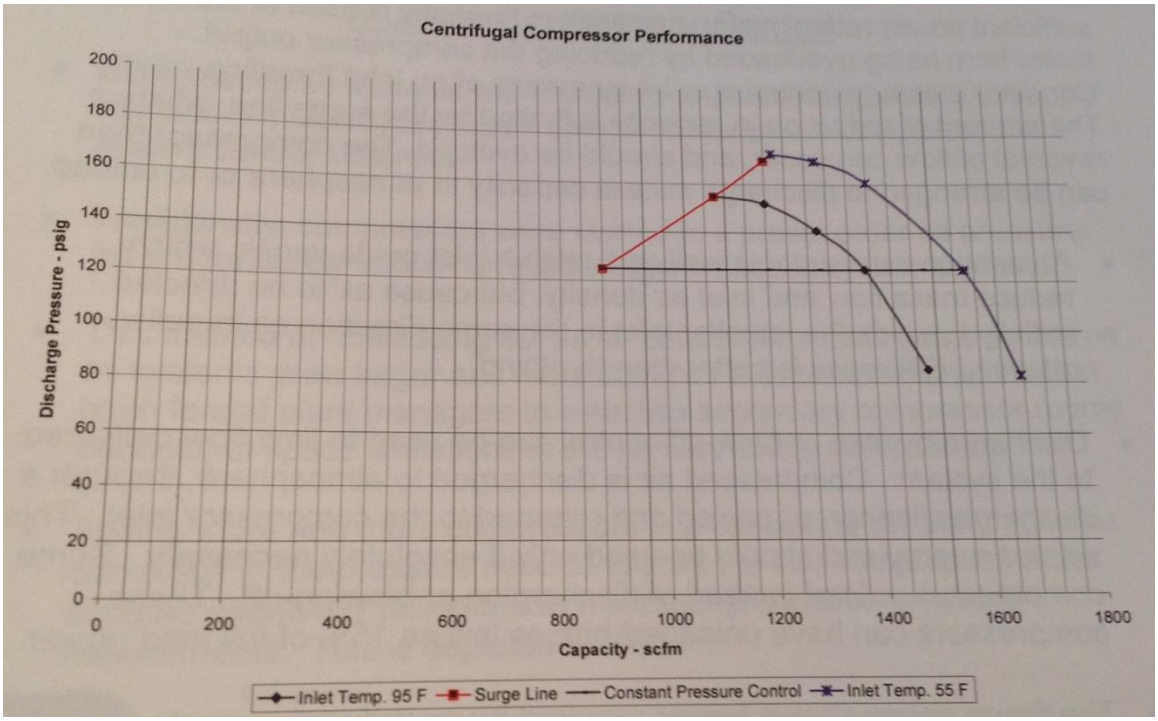


Figure 22: The Effect of Inlet Air Temperature (11)

3.3 Multiple Compressor Control

Multiple compressor systems are quite common in larger facilities. It is of paramount importance that multiple compressors are controlled to ensure compressors are not operating when they are not needed. As more compressors are added to a system, the complexity of the control scheme increases.

3.3.1 Cascade Control

Traditionally, cascading controls were used to start compressors in a predetermined order as compressed air demand increases and system pressure falls. To get a sense of what is happening with one compressor, the unload set point and the full load set point will be discussed. As the pressure in the system increases and exceeds a compressors set point, the compressor will unload to save energy. If the system pressure falls below the lower pressure set point after the compressor unloads, the compressor will reload (11). Cascade control is illustrated in Figure 23 below.

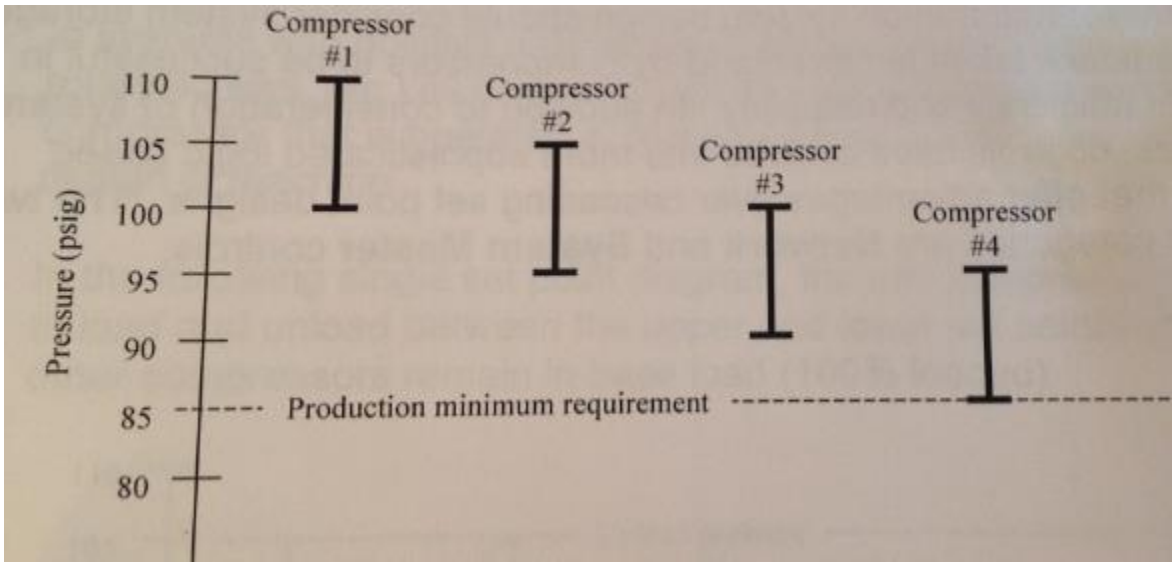


Figure 23: Compressors in Cascade (11)

The top of the each bar represents the pressure at which the compressor unloads. Conversely, the bottom of each bar represents the pressure at which the compressor is fully loaded. The

issue with this type of control scheme is that the last compressor in the cascade will potentially allow the system pressure to dip below the production minimum requirement. There is always a lag between when a compressor starts up and when it begins to deliver compressed air, thus the system pressure could fall below the minimum pressure before demand is met (11). To avoid this problem, a facility employee may simply disable the control system, or notch up the pressure band of each compressor, which leads to inefficiency. Tighter overlapping pressure bands may also be chosen as a solution, but this will cause unnecessary starts, leading to the purchase of another compressor. The real issue with cascade systems is that time is rarely considered. If enough storage is installed, the compressors will have plenty of time to react to demand changes. Instead of the purchase of an additional compressor, which will likely only match demand efficiently 15% of operation, storage should be considered. Even more complex control systems exist for controlling multiple compressor systems. Mainly, this consists of Network and System Master Controls (11).

3.3.2 Network Controls

Network controls are used for larger systems of compressors, and are better suited for avoiding part loading of compressors. Network controls use the already existing control microprocessors to link together multiple compressors. This forms a chain of communication that makes it easier to decide when to stop/start, load/unload, modulated, vary displacement, or vary speed for a compressor. Generally, in a system of compressors, one compressor is the

lead compressor, which operates constantly. Other compressors in such a system would be subordinate to the demands of the lead compressor.

Traditionally, network controls will have all necessary compressors, except one, fully loaded. The compressor that is not fully loaded is the trim compressor, which is operated partially loaded to meet fluctuations in demand (11).

The system can be dynamic, in that as pressure increases to a point above the unload pressure or below the load pressure changes in the system operating can be made. For example, if the system pressure increases even after the trim compressor unloads, one of the base load compressors will begin to unload. When this happens, the system pressure will likely begin to fall and the former base loaded compressor can begin to trim. The former trim compressor can be shut down after a set run time and cool-down timers are finished. The former trim compressor will turn back on and continue to trim during high demand periods (11).

The pressure sensor is typically downstream from the lead compressor to a central point where all compressed air meets (11). An example of network controlled system can be viewed in Figure 24.

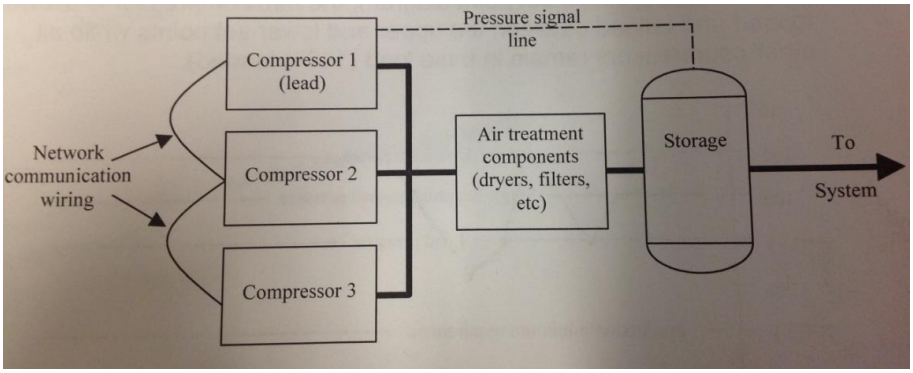


Figure 24: Network Controls

There are potential pitfalls with network controls. Using a pressure downstream of air treatment equipment could result in higher compressor discharge pressure due to increased pressure drop over time through equipment. Measures must be made to ensure compressors are not compressing air above maximum allowable discharge pressures. Typically, network controls only work with compressors of the same brand, and cannot be networked with remote compressor rooms. Also, there is no method of networking positive displacement and dynamic compressors. System master controls are required for more complex compressor systems such as these.

3.3.3 System Master Controls

For facilities with complex compressed air systems, consisting of both positive displacement and dynamic compressors, and remote compressor rooms, system master controls can safeguard against compressors coming online when they are not needed. In addition to running a more efficient compressed air system, such as monitoring and controlling all

components in the system, system master controls can trend data to better help with maintenance, thereby reducing overall operation costs (11).

The simplest system master controls will utilize cascading set point logic to control air compressors within the system. High-tech system master controls utilize a technique called single point control logic. This uses rate of change dynamic analysis to decide how compressors will react in regard to changes, whether it be from the supply side, demand side, or atmospheric conditions. Compressor demand is influenced by what are termed end use-events. Events influence system demand either positively or negatively, and the control system must react accordingly (11). A few examples of events include shift change, line purge, dense phase transport, and compressor failure (11).

A system master control can perform many different and complex functions. However, the number of functions a particular system will have is given to practicality and cost. For example, if the dewpoint of the compressed air must be controlled to a tight band, it would make sense to install a sensor after the air dryer to communicate with the System Master control. Some examples of the possible functions of a System Master control are; send/receive communications, communicate with plant information systems, monitor weather conditions, adjust pressure/flow controller set points, monitor filter differential pressure, start/stop and load/unload compressors, change base/trim duties, and select the appropriate mixture of compressors to optimize efficiency (11).The purchase of sensors may cost as little

as $300, or as high as $1,500 depending on the application (11). Another potentially important controller is a pressure/flow controller.

3.3.4 Pressure/Flow Controllers

If a facility requires tight pressure bands for production, a Pressure/Flow controller might be considered. Typically, in a multi-compressor system, the multiple pressure control bandwidths will overlap, which could cause large variance in pressure. Also, facilities with only one modulating rotary screw compressor will have an approximate pressure band of 3 to 10 psi, which may be undesirable. A pressure/flow controller will control the pressure and flow coming from a single compressor or a multiple room compressor system and drop the pressure to the desired facility pressure. It is an excellent way of setting which compressor room will have base compressor duties, and which will have trim duties. For example, in a two compressor room system with outputs of 125 psig and 100 psig respectively, the Pressure/Flow controller can allow the 100 psig room to be the base, and have the 125 psig room trim while decreasing pressure to a standard facility pressure (11).

Pressure/flow controllers respond very quickly to demand events, which allows for stable pressure output. The pressure flow controller will respond much quicker to demand events than standard compressor controls. With properly sized storage upstream, the pressure flow controller will react quickly enough to a demand event to avoid the need to start another air

compressor. Below is a schematic of a pressure/flow controller in a single room compressed air system.

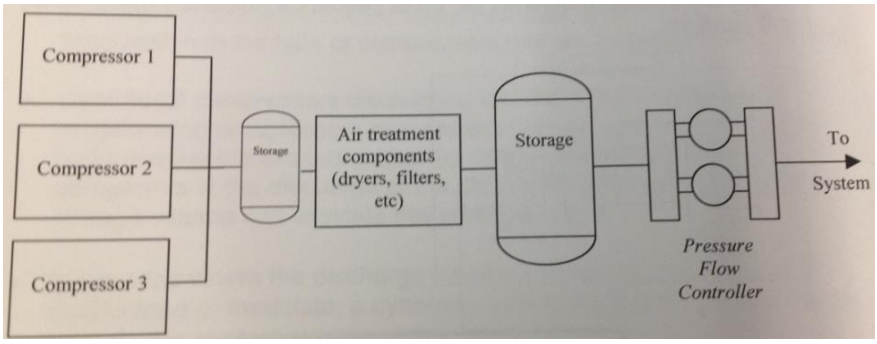


Figure 25: Pressure Flow Controller with One Compressor Room (11)

Pressure/flow controllers can be vital to operating a compressed air system with both positive displacement and dynamic compressors. A centrifugal compressor's performance can be influenced in several ways by positive displacement compressors with varying discharge pressure. An example of this influence is when a load/unload or modulating compressor's pressure band overlaps the throttling line of the centrifugal compressor, the centrifugal will blow off any excess compressed air to atmosphere. Additionally, if a multiple compressor system cascades compressors, and system pressure drops as compressors come online, a centrifugal compressor will begin to operate in a condition known as stonewall. This occurs when the compressor experiences less than design pressure at high flow rate. The compressor pressure will plummet and lose the ability to produce additional flow. While this is not harmful to the compressor, the compressor does not realize its full potential. To avoid these

issues, a pressure/flow controller can operate in back-pressure mode for the dynamic compressor, while operating with a forward pressure/flow controller for the positive displacement side of the system. The back pressure pressure/flow controller has a pressure set point above the choke point and below the set point of the dynamic compressor, which ensures the dynamic compressor will not be subject to surge or choke (11).

Chapter 4 – Data Collection

The subject facility is a textile manufacturer in central North Carolina. Large volumes of compressed air are used throughout the manufacturing of their high quality product. The facility's compressed air system consists of four compressors flooded oil screw compressors. Compressor 1 is a 500 hp Sullair TS-32, compressor 2 is a 600 hp Sullair TS-32, compressor 3 is a Sullair 150 hp rotary screw utilizing modulating controls, and the final compressor is a Sullair 100 hp rotary screw utilizing load/unload controls. The table below lists the facility air compressors.

Table 2: Facility Compressors

Compressor #	Model #	Voltage	Rated Pressure (psig)	Rated Power (hp)	Rated Volumetric Flow
1	TS-32S-500	460	100/150	500	2,530 ACFM
2	TS-32S-600	460	100/150	600	3,000 ACFM
3	20-150	460	100/110	150	750 CFM
4	20-100l	460	100/110	100	500 CFM

To gain a better understanding of how the system is operating, collection of pertinent data is required. Parameters required for proper analysis are the compressor power trending over time for each compressor, pressure trending over time, and total system capacity, in cubic feet per minute, over time. The facility monitors compressed air flow continuously, but not

amperage to the compressors or system pressure. Thus, installation of data loggers is required to acquire amperage and pressure readings. As a note about the facility, the facility uses compressed air 24 hours per day, but typically does not require compressed air on Sundays.

4.1 Measurement Equipment

Onset Hobo data loggers and Onset HOBOware software is used to collect data regarding amperage and system pressure. The H22 data logger was chosen for this project. The H22 system consists of a HOBO H22 Data Logger, a FlexSmart TRMS Module, and the necessary HOBOware Pro Software. Additionally, a current transducer and a pressure logger will be required for data collection. These items are depicted in the figures below.

Figure 26: FlexSmart TRMS Module

The FlexSmart Analog Module is a DC signal-conditioning module for the H22 HOBO Energy Logger. It is equipped with two channels, which accepts a wide range of sensors and transducers. The next item is the logger itself, the HOBO H22 Data Logger, which is a 15-channel system. The H22 can monitor many different measurement types, including gauge pressure, AC current, and compressed air flow. Figure 27 below depicts the HOBO H22 Data Logger.

Figure 27: HOBO Energy Logger

The modules connect to one of the three ports, and the sensor to record measurements connects to the module by means of a detachable screw terminal connector. For this study, 200 and 1500 amp current transducers were used to monitor AC current flowing to the electric motors of the compressors. A CT senses voltage through a wire, and then scales it to a matching amperage. Figure 28 is a close up of a 200 amp Current Transducer (CT).

Figure 28: 200 AMP Current Transducer

Figure 28 shows the CT connected to the FlexSmart module, which is connected to the HOBO H22 Energy Logger. Before data logging can commence, the logger must be programmed and launched.

Figure 29: Data Logger Setup

To launch the data logger, one must first open the HOBOware program and connect the data logger to the computer using an interface cable. When the program is started, the user can launch the device by clicking Launch under the Device tab. This is shown in Figure 29.

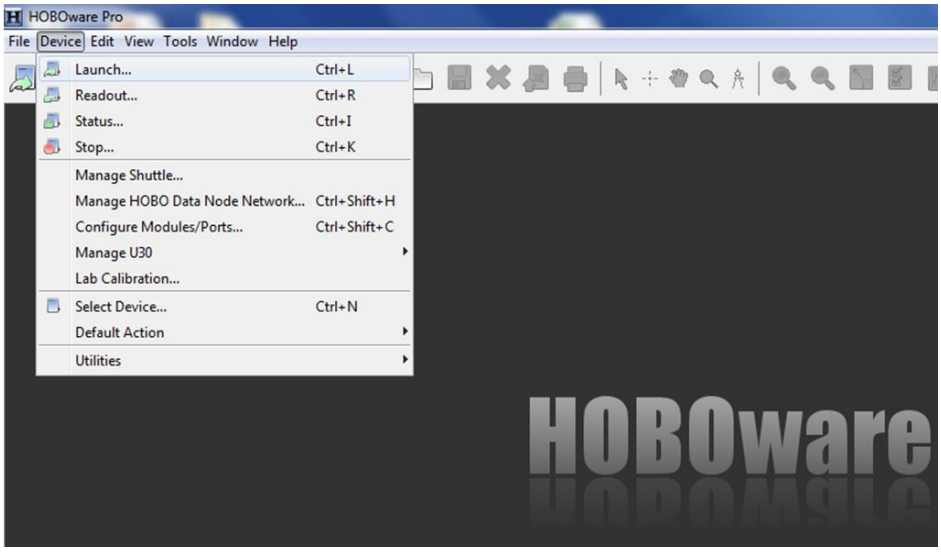


Figure 30: Launching a Device

HOBOware then recognizes the device to be launched. By moving forward, the data logger can be programmed.

The Launch Logger screen allows the user to add a description, configure sensors, and select a launch time. If logging a compressor, the description may simply be the number of a compressor in a system of air compressors. Choosing the required sensors for logging is also required. The most pertinent data needed when logging an air compressor is the amperage, so this may be selected. Figure 31 displays this interface.

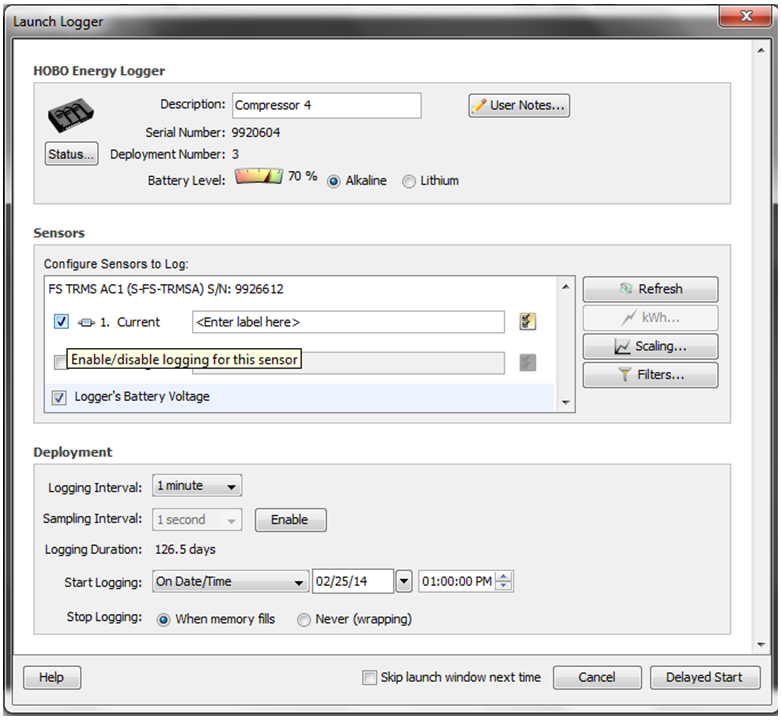


Figure 31: Launch Logger Interface

After the sensors to log are selected, they must be scaled to match the CT. For example, if the CT measures 0 mV, the corresponding amperage would be 0. Likewise, if a CT measures a maximum of 333 mV, this would correspond to the maximum amperage that CT is valid for. This means that for a 200 amp CT, if the CT measures 333 mV the amperage through the wire is 200 amps. These points must be configured to accurately measure amperage. The interface to scale parameters is shown below in Figure 32.

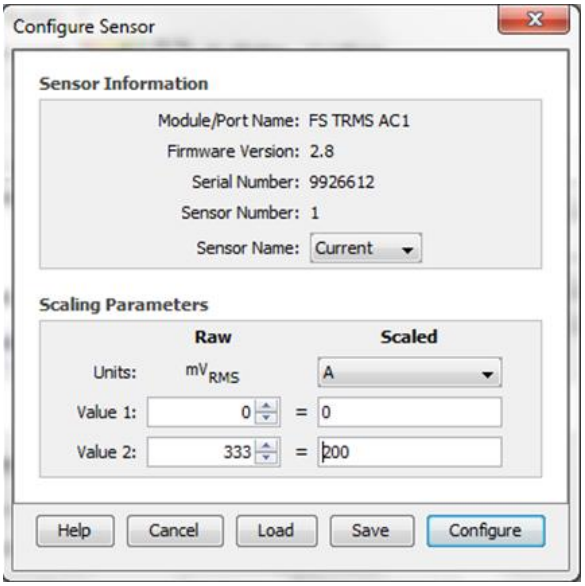


Figure 32: Configure Sensor

After configuration, the launch time is set. There are two options; the first is to simple launch the logger immediately, but if multiple loggers are being used to monitor a compressed air system, a delayed launch time can be chosen for all loggers. Because this project dealt with four compressors and it is ideal for all loggers to begin logging at the same time, delayed start was used.

4.2 Measurement Procedure and Data Collection

The initial phase in the experimental analysis of the compressed air system operation involved collecting data regarding the amperage draw of each compressor, the system pressure, and system volumetric flow rate. There were two separate visits to collect two sets of data.

An initial facility visit on January 14th, 2014 was setup to become familiar with the compressed air system, and program data loggers to monitor amperage and system pressure. The data loggers for amperage readings were programmed to record a reading every 5 seconds, whereas the pressure transducer recorded readings every minute. For the 100 hp and 150 hp air compressors, 200 amp current transducers were used, whereas 1,500 amp current transducers were utilized for the larger 600 hp and 500 hp compressors.

To indicate the air compressor to which each data logger would be installed, a specific label was attached to each data logger. The data loggers were left with the maintenance manager to be installed. These loggers remained logging until January 30th 2014, for a total of 17 days of data collection. On January 30th 2014 the loggers were retrieved to analyze the data. Additionally, volumetric flow data for this time period was requested for a complete overview of the system operation. Unfortunately, the total volumetric flow was unavailable for this time period, thus a second data collection period was set up and initiated to monitor amperage and pressure. Additionally, volumetric flow data was requested for this second period of data collection.

On February 11th, 2014 the five programmed data loggers for the compressor and pressure monitoring were dropped off at the facility to be installed by the maintenance manager. The data loggers collected data from the 11th of February to the 27th of February. Volumetric flow data from February 21st to February 28th was provided by the facility. The flow data was

recorded every 30 minutes. Upon viewing the data from February 11th to February 27th, it was determined that the data logger monitoring system pressure malfunctioned and did not record the pressure. Therefore, it was decided to use the first set of data for analysis and to estimate a volumetric flow for this period.

4.3 Data Analysis

The amperage was recorded every 5 seconds, but to analyze multiple compressors at once, the data was averaged over 1 minute periods. This means that 12 five second amperage recordings were averaged into one data point.

First, the compressor horsepower for each compressor is plotted, and then the power for each compressor, pressure, and volumetric flow will be integrated. To calculate the compressor power in kW, the following formula is used:

$$Power(kW) = \frac{Amperage \times Voltage \times \sqrt{3} \times PowerFactor}{1000} \quad \textbf{(Equation 1)}$$

The voltage to the compressor is 460 volts. The electric motors in each of the air compressors is three phase. To convert kW to brake horsepower the following formula is used:

$$Power(hp) = \frac{kW}{0.746\frac{kW}{hp}} \times MotorEfficiency \quad \textbf{(Equation 2)}$$

Power factor varies depending on percent load on the compressor motor. An analysis to determine the power factor at percent of full load amps follows. The motor efficiency

depends on the size of motor, and is fairly constant for motors that are at least 50% loaded. For this study we assumed the motor efficiency to be that of a standard efficiency motor. These values are taken from AIRMaster+ modeled compressors for consistency throughout the study. All subsequent horsepower values are in brake horsepower.

Table 3: Motor Efficiencies

Compressor #	Model #	Rated Power (hp)	Motor Efficiency
1	TS-32S-500	500	94.5%
2	TS-32S-600	600	94.5%
3	20-150	150	93.0%
4	20-100l	100	91.7%

4.3.1 – Power Factor Analysis

Depending on the loading of the compressor, the motor power factor can vary between 0.3 and 1.0. This would greatly reduce the compressor power during periods of unloading, despite the current remaining high. Thus, it is especially important to quantify an accurate power factor for compressors that utilize load/unload controls, although the power factor may vary slightly for a compressor uses modulation. The Department of Energy software program AIRMaster+, which will be used to model the compressed air system to determine and quantify efficiency measures, has a feature that determines the calculated power of an electric motor at specific amperages. The calculation assumes a power factor and calculates the

power in kW. See the AIRMaster+ power calculator in Figure 33 below. The voltage used in the calculator is 460 volts, and the electric motor is three phase.

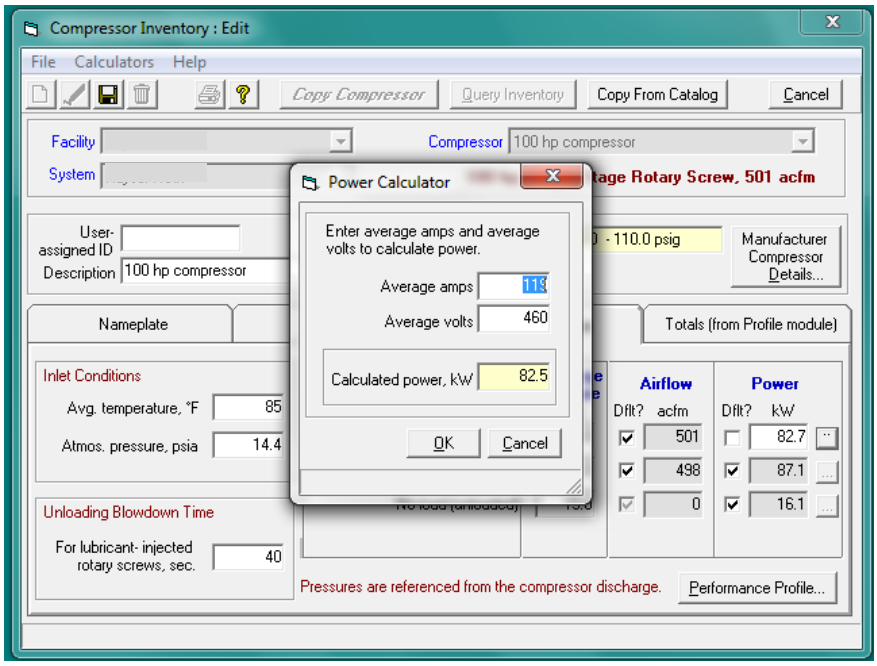


Figure 33: AIRMaster+ Power Calculator

Using the 100 hp compressor, the next step is to tabulate power calculations starting with 0 amps all the way to 135 amps, which is the full load amperage. The power is then calculated with a power factor of 1. To calculate the power factor the AIRMaster+ power calculations are divided by the power with a power factor of one. The results are listed in Table 4.

Table 4: 100 hp Compressor Power Factor

Amperage Input	Air Master Power, kW	Power PF = 1 kW	AIRMaster+ Power Factor	Trend Line Power Factor Calculated	Power kW	Power hp
0	0	0	0	0	0	0
5	1	3.98	0.25	0.20	0.8	1.08
10	2.1	7.97	0.26	0.28	2.2	2.9
20	4.2	15.93	0.26	0.27	4.3	5.7
30	6.2	23.90	0.26	0.25	6.0	8.06
40	8.3	31.87	0.26	0.33	10.5	14.0
50	21.8	39.84	0.55	0.49	19.5	26.1
60	34.2	47.80	0.71	0.67	32.1	43.0
70	44.5	55.77	0.78	0.81	45.4	60.8
80	53.5	63.74	0.84	0.88	56.2	75.3
90	61.6	71.71	0.86	0.88	63.1	84.5
100	69.1	79.67	0.87	0.85	67.4	90.3
110	76.2	87.64	0.87	0.83	72.9	97.8
120	82.5	95.61	0.86	0.87	82.8	110.9
125	86.6	99.59	0.87	0.87	86.7	116.1
130	90.1	103.58	0.87	0.87	90.1	120.8
135	93.6	107.56	0.87	0.87	93.6	125.4

The AIRMaster+ power factor is graphed against amperage input, and a polynomial trend line is created. See Figure 34 for the plot of power factor against amperage. The trend line resulted in the following equation:

$$PF = -2E-11x^6 + 1E-08x^5 - 2E-06x^4 + 0.0001x^3 - 0.0041x^2 + 0.0586x \quad \text{(Equation 3)}$$

Above 90 amps the power factor levels out to approximately 0.87. This is reflected in the Trend Line Power Factor Calculated column in Table 4 above.

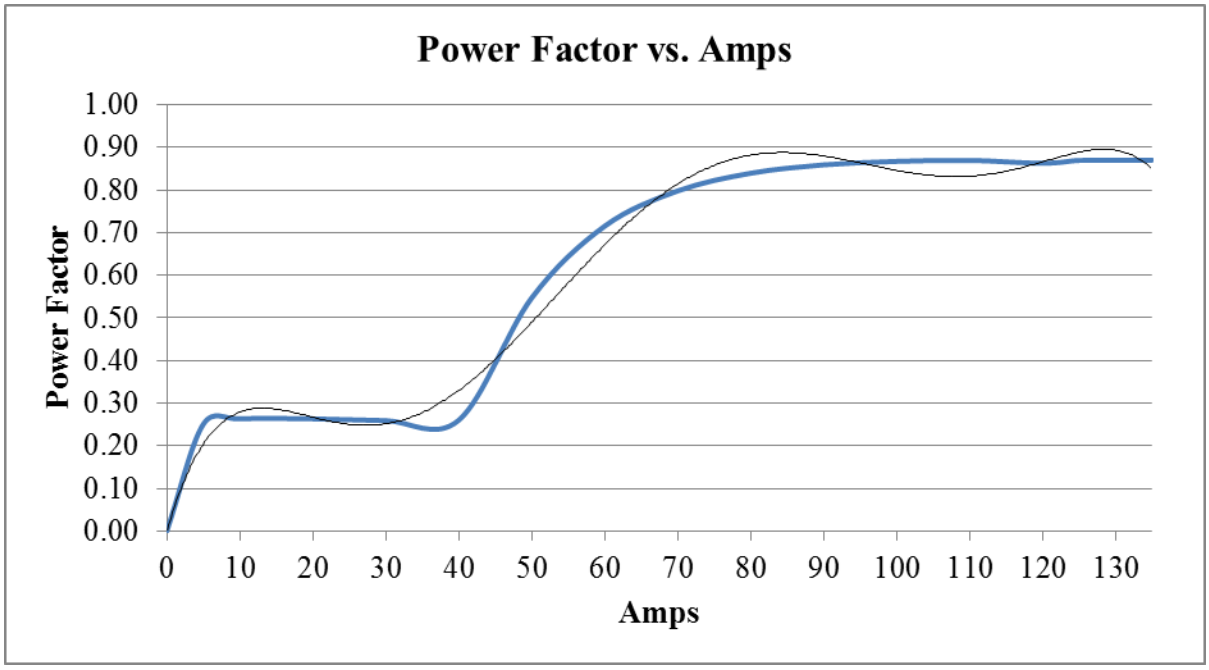


Figure 34: Power Factor vs. Amperage

To calculate a corrected power factor for all compressors, percent full load amperage is plotted against the AIRMaster+ calculated power factor. A trend line equation is computed for calculating power factor for a percent of full load amperage for any compressor. Each amperage input from Table 4 is divided by 135 amps to calculate percent full load amperage, and the corresponding AIRMaster+ power factor is carried over.

When using Excel to calculate power factor with the trend line equation, an IF statement is used to qualify everything above 88% full load amperage to equal a power factor of 0.87. See Table 5 for power factor values at corresponding percent full load amperages.

Table 5: % Full Load Amperage and Power Factor

%Full Load Amperage	AIRMaster+ Power Factor	Trend Line Power Factor Calculated
0.00	0.00	0.00
3.70	0.25	0.20
7.41	0.26	0.28
14.81	0.26	0.27
22.22	0.26	0.25
29.63	0.26	0.33
37.04	0.55	0.49
44.44	0.72	0.67
51.85	0.80	0.81
59.26	0.84	0.88
66.67	0.86	0.88
74.07	0.87	0.85
81.48	0.87	0.83
88.89	0.86	0.87
92.59	0.87	0.87
96.30	0.87	0.87
100.00	0.87	0.87

The power factor calculated using the trend line equation is fairly close to the AIRMaster+ power factor calculations; therefore it is reasonable to use the trend line equation in subsequent analysis. The trend line is 6th order polynomial. The trend line equation is given by:

$$PF = -1.4059088822E\text{-}10x^6 + 4.5145915214E\text{-}08x^5 - 5.4563692241E\text{-}06x^4 + 3.0324285292E\text{-}04x^3 - 7.5529979405E\text{-}03x^2 + 7.9049331517E\text{-}02x$$ **(Equation 4)**

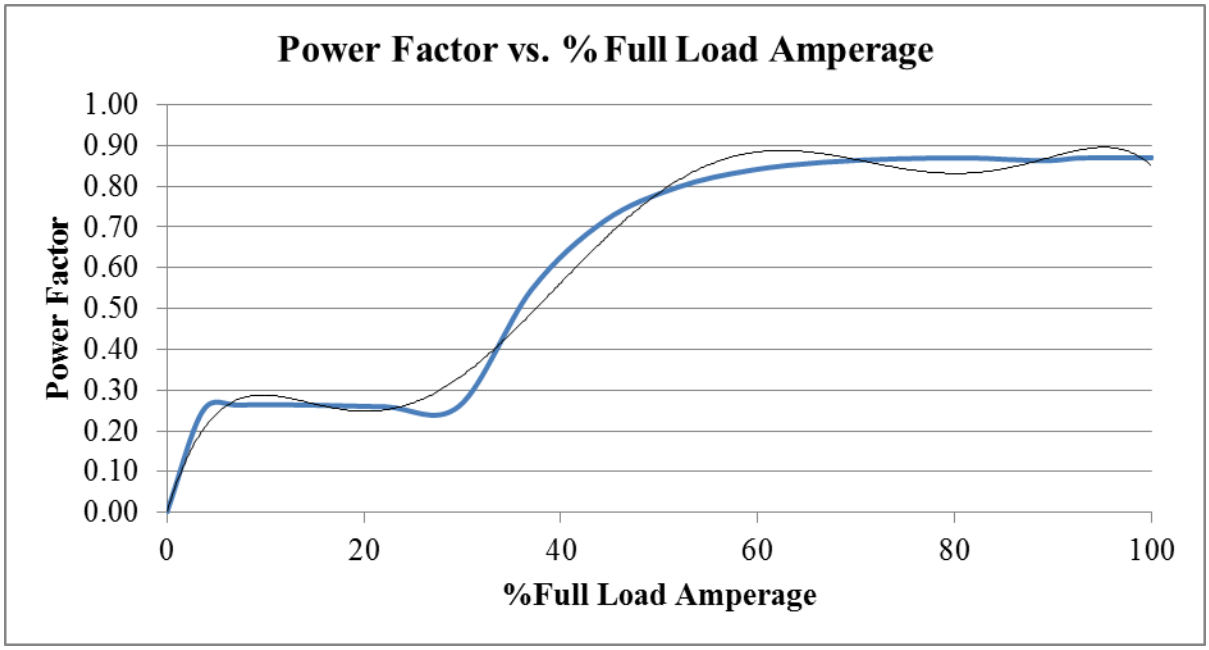


Figure 35: Power Factor vs. Percent Full Load Amperage

When calculating compressor power, the collected amperage data will first be divided by the full load amperage to calculate percent full load amperage. Next, Equation 4 is used to calculate an estimated power factor. This power factor will be used to calculate power in kW, which is then converted to horsepower. This method is used to calculate power for each of the compressors in the subsequent sections.

4.3.2 - 100 hp Air Compressor Data Analysis

Below, in Figure 36 is a plot of the 100 hp compressor amperage from January 14th to January 30th. This data comes from the raw amperage data, which was recorded every 5 seconds.

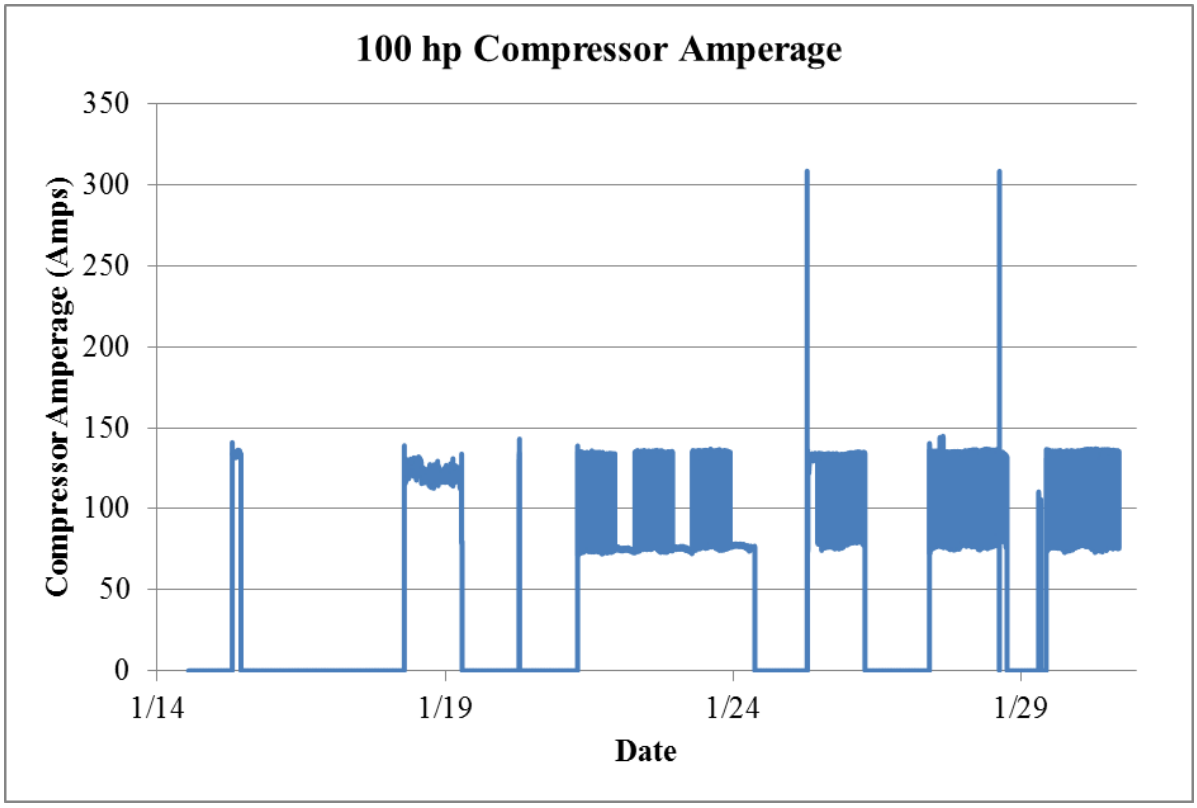


Figure 36: 100 hp Compressor Amps from Raw Data

The compressor is utilizing load unload controls, which is seen clearly during the period of January 21st to January 24th. The spikes in the data, in which the compressor power spikes close to 300 amps, come from when the compressor first starts up. This likely caused by locked-rotor current, which occurs due to high initial inductance, requiring high amperage to turn the motor. During the data collection period, the average power from this compressor was 47 hp. Figure 37 is a plot of the 100 hp compressor power using averaged data points. Averaging data points together will smooth out the data, thereby precluding high amperage

readings from locked rotor amperage. The power factor is calculated based on the percent full load amperage for each data point.

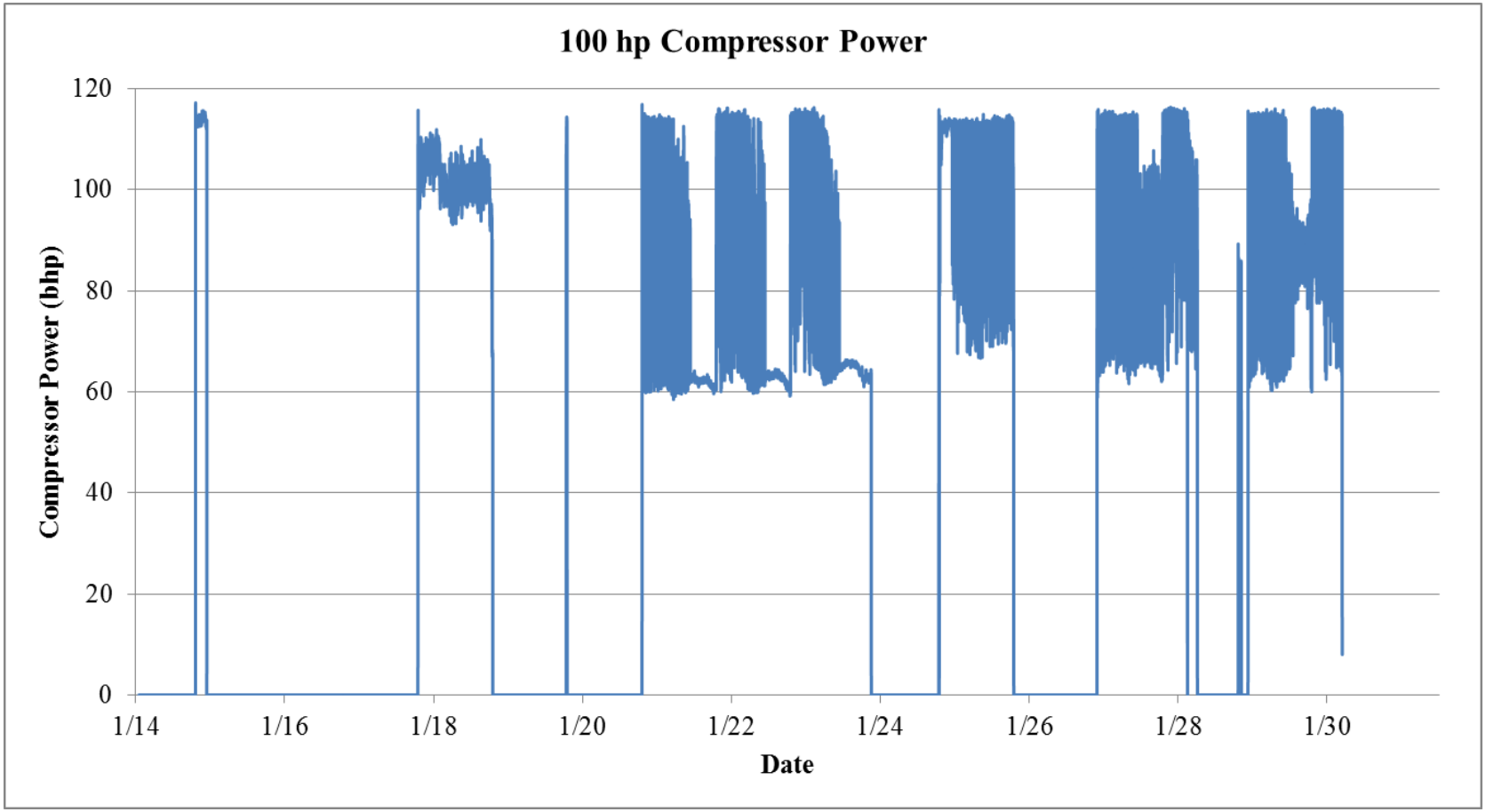


Figure 37: 100 hp Compressor Power from Averaged Data Points

The following Figure details the load/unload cycle of the 100 hp compressor.

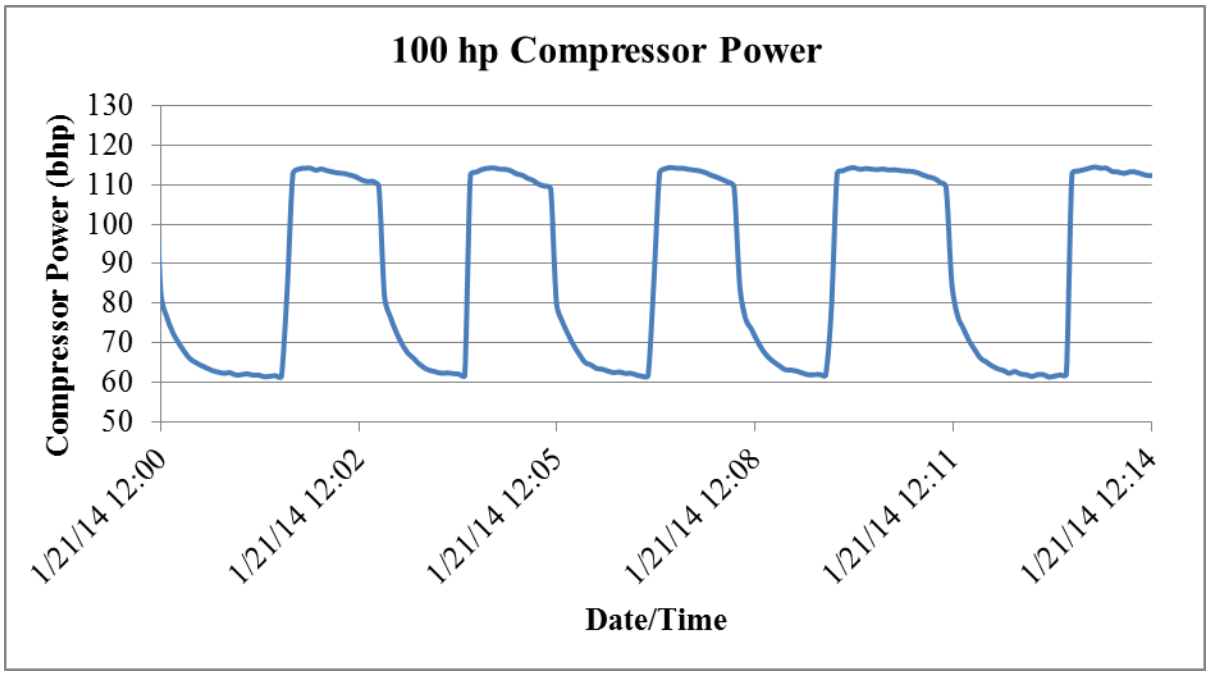


Figure 38: 100 hp Compressor Load/Unload Cycle

During the load/unload cycle, the maximum power draw is 115 hp and the minimum power is 60 hp. The compressor unloads for approximately 1.5 minutes, gradually reducing the power input. The compressor unloads to 60 hp for little time, almost immediately reloading to full power. This operation is not ideal, as you would like to see the compressor remain unloaded for as much time as possible. Unloading for greater periods of time would lower the average compressor power during operation. Currently, during the load/unload cycle; the average compressor power is 85 hp.

4.3.3 – 150 hp Air Compressor Data Analysis

Figure 39 below depicts the amperage to the 150 hp compressor during the period of January 14th to January 30th.

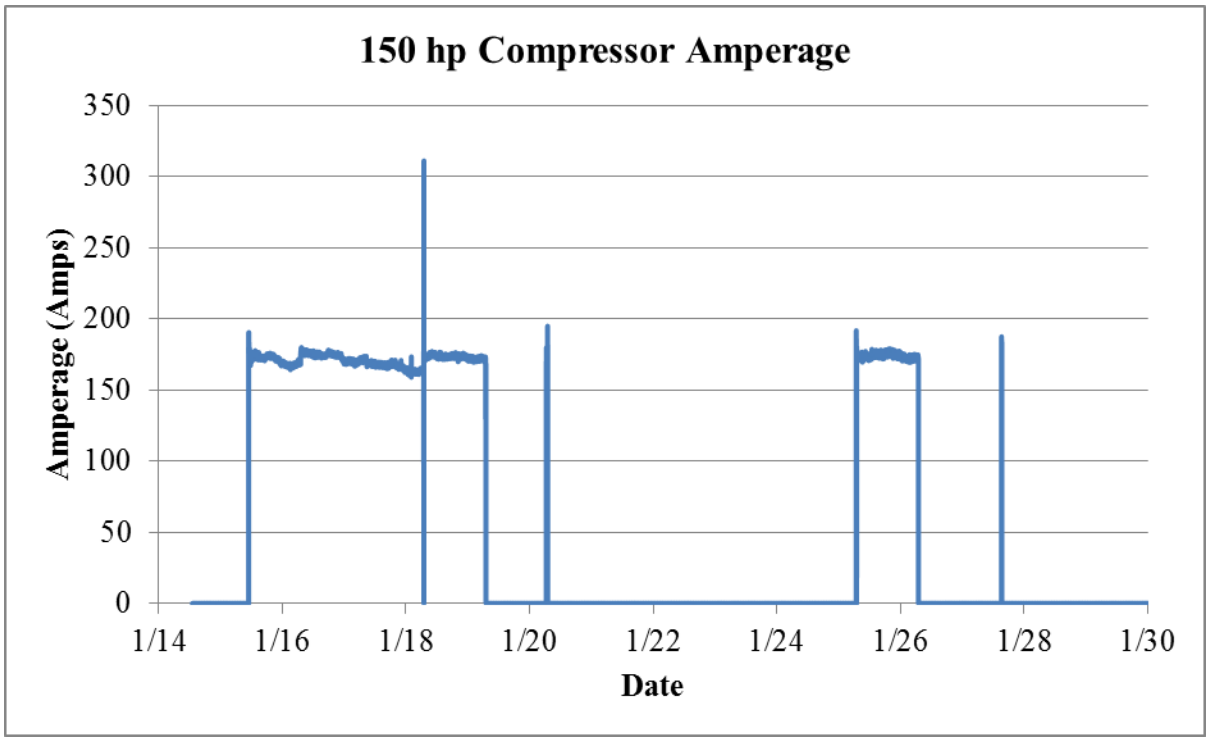


Figure 39: 150 hp Compressor Amperage

Figure 40 below is a plot of the 150 hp compressor power. The 150 hp compressor is utilizing modulating controls, and uses an average of 150 hp while producing compressed air. During the logging period from January 14th to January 30th, the compressor had an average power of 44 hp. The compressor does not illustrate the same load/unload cycle as the 100 hp

compressor, which is depicted in Figure 41 below. The compressor clearly responds quickly to changes in compressed air demand through modulating controls.

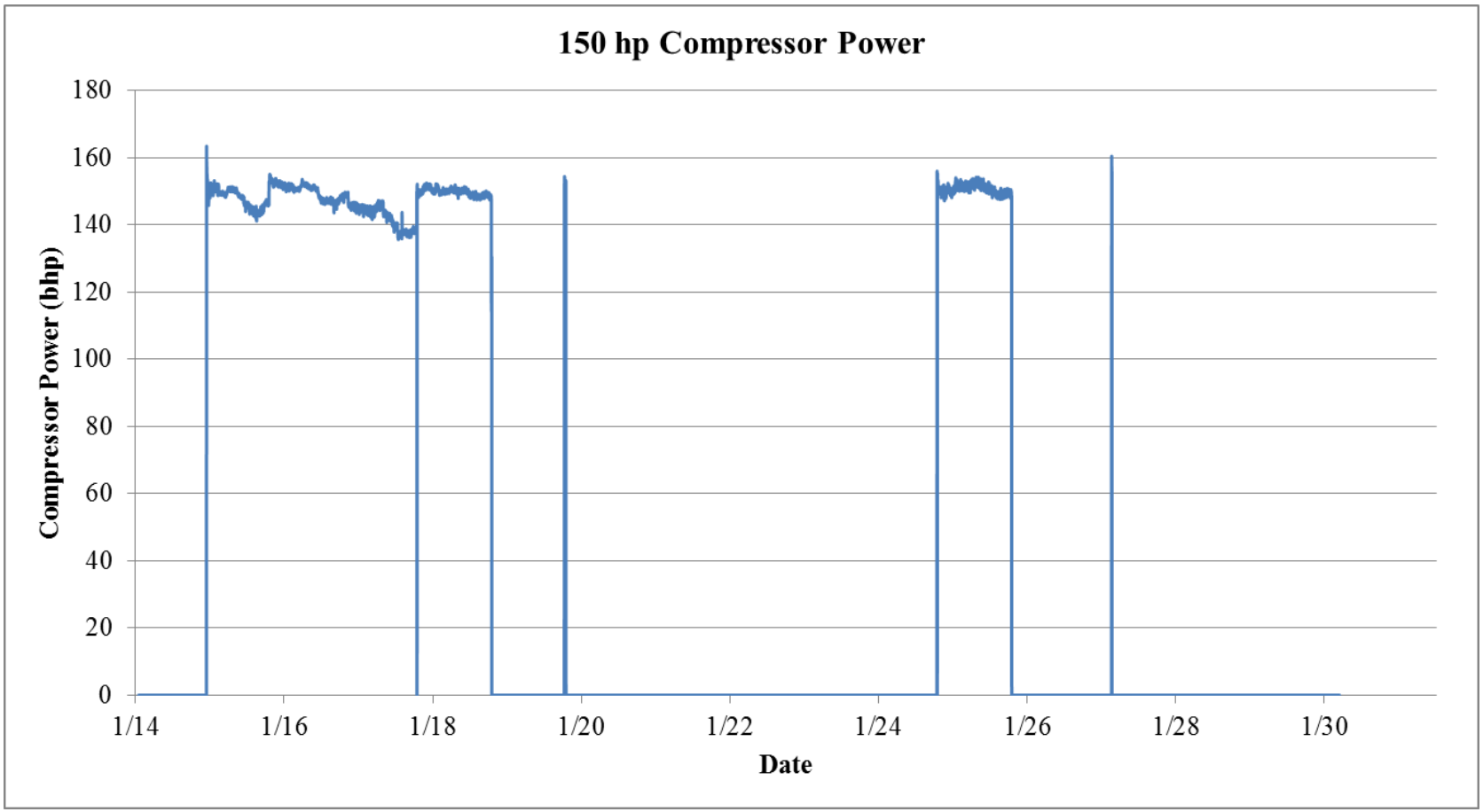


Figure 40: 150 hp Compressor Power

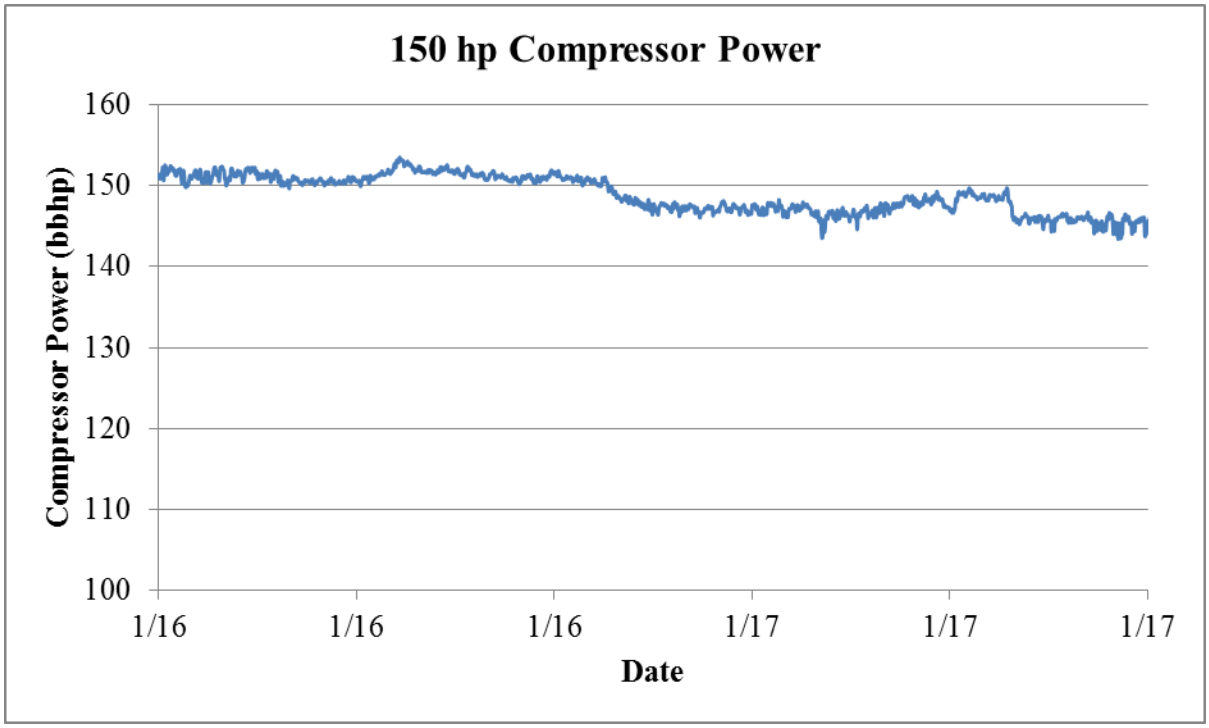


Figure 41: Illustration of Modulation

4.3.4 – 500 hp Air Compressor Data Analysis

The larger air compressors supply much of the compressed air demand for the facility. The TS-32S 500 hp compressor produces 2,530 ACFM at full load power. The amperage flowing to this compressor over two and a half week data logging period is graphed below in Figure 42.

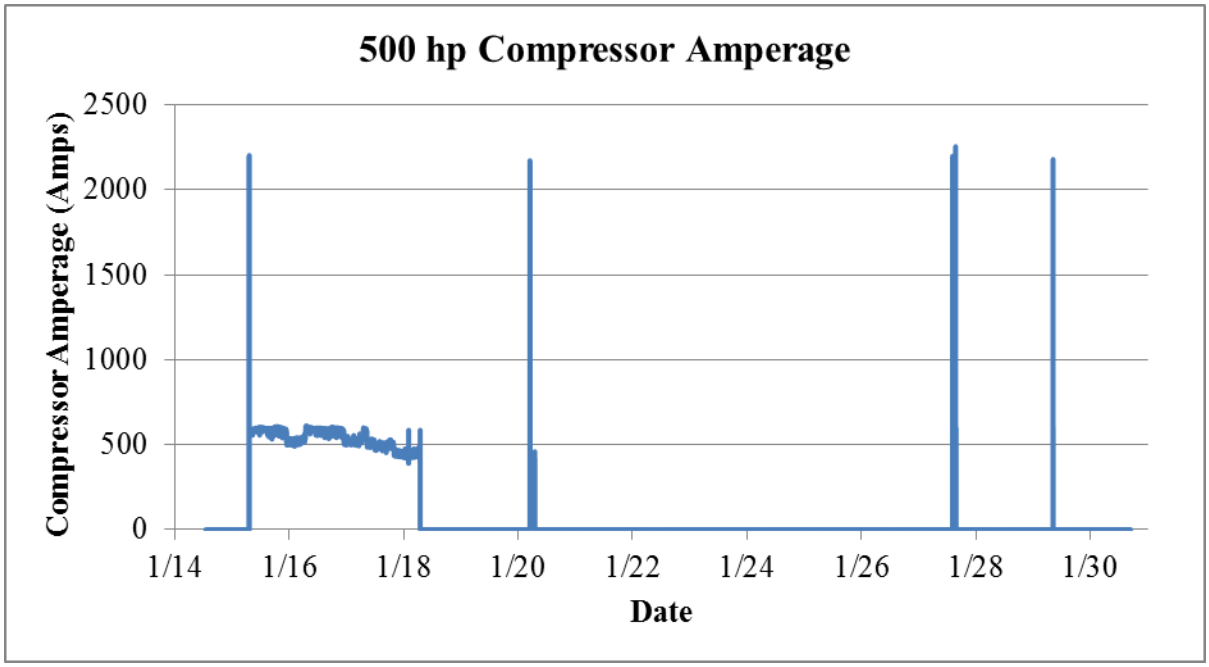


Figure 42: 500 hp Compressor Amperage

Interestingly we see large spikes of current, which to indicate the amperage to the electric motor power to be above 2,000 amps. The compressor turns on, has a large inrush of current, and then almost immediately turns off. This occurrence should not happen, as a compressor should only turn on when it is absolutely needed. It is likely arising when the facility first starts up, and the compressors must pressurize the system for production. At that moment, the system turns all compressors on to fulfill the demand to pressurize the system. However, this may incur a large billing demand. Thus, a solution to this issue might be to control which compressors turn on while pressurizing the system.

The compressor has an average power of 87 hp during the entire logging period, and an average of 462 hp while producing compressed air. Figure 43 is a plot of the compressor power utilizing averaged amperage data to smooth out the data and exclude large spikes in amperage during initial compressor startup.

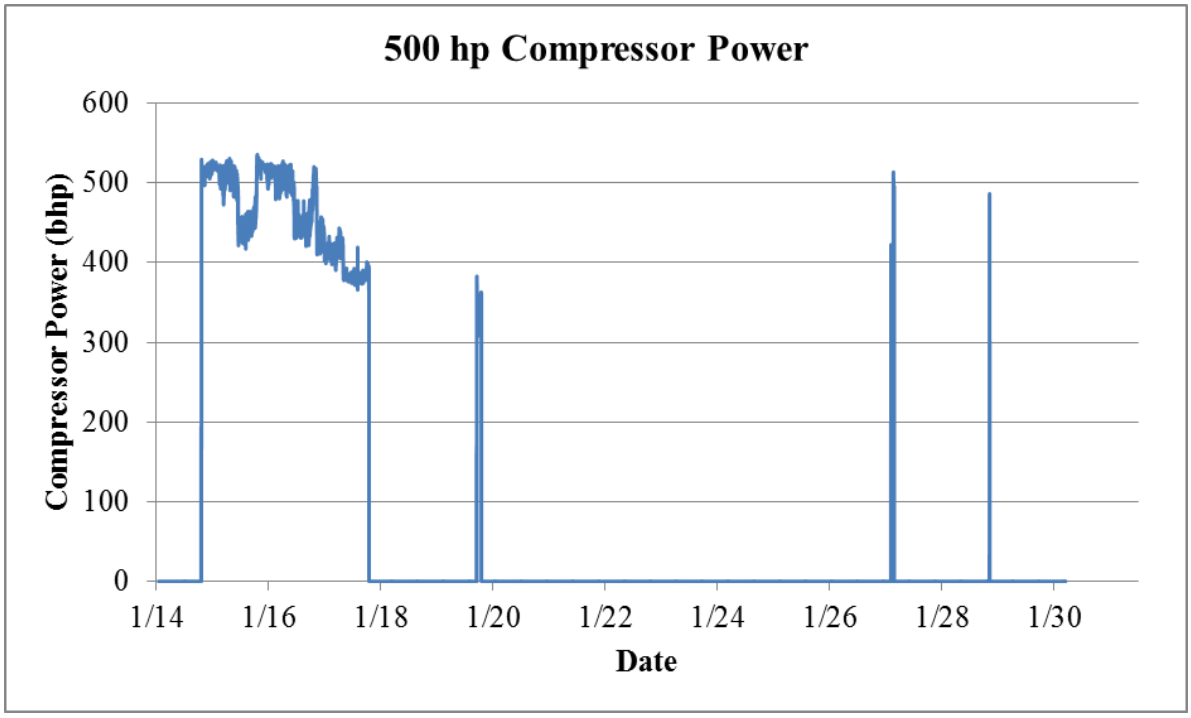


Figure 43: 500 hp Compressor Power

4.3.5 – 600 hp Air Compressor Data Analysis

The 600 hp compressor is the primary compressor, which satisfies the greater part of the compressed air demand when in operation. The compressor operates with modulating controls to match compressed air demand. During the data collection period, the compressor

had an overall average power of 317 hp. While the compressor was in operation, the average power draw was 601 hp. The large spike in the amperage is likely due to locked rotor amperage during the initial startup of the electric motor. This can be seen in Figure 44 below. Figure 45 focuses and depicts the 600 hp compressor's power during the logging period of January 14th to January 30th. The 600 hp compressor typically supplies the majority of the compressed air demand; therefore the 500 hp compressor does not operate in conjunction with the 600 hp compressor. However, both the 500 hp and 600 hp compressor turn on when the facility initiates startup after a weekend. When this occurs, the 500 hp compressor almost immediately shuts off and the 600 hp air compressor remains on.

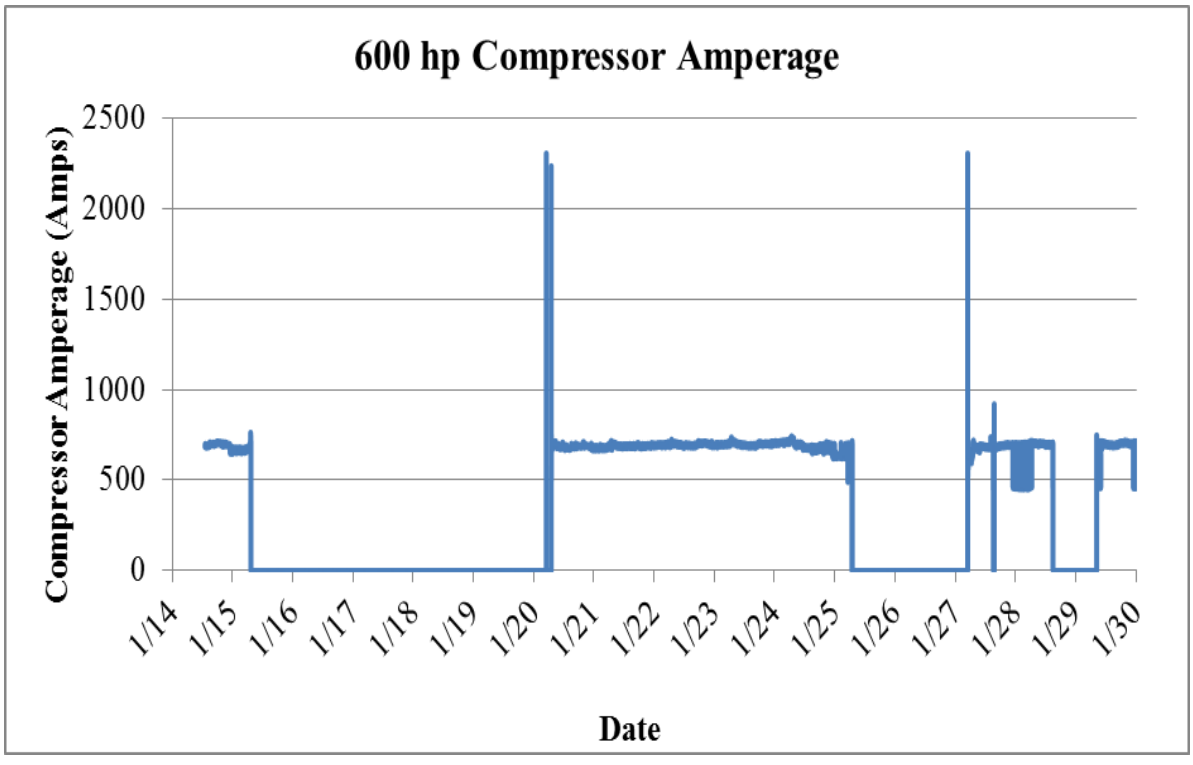


Figure 44: 600 hp Compressor Amps

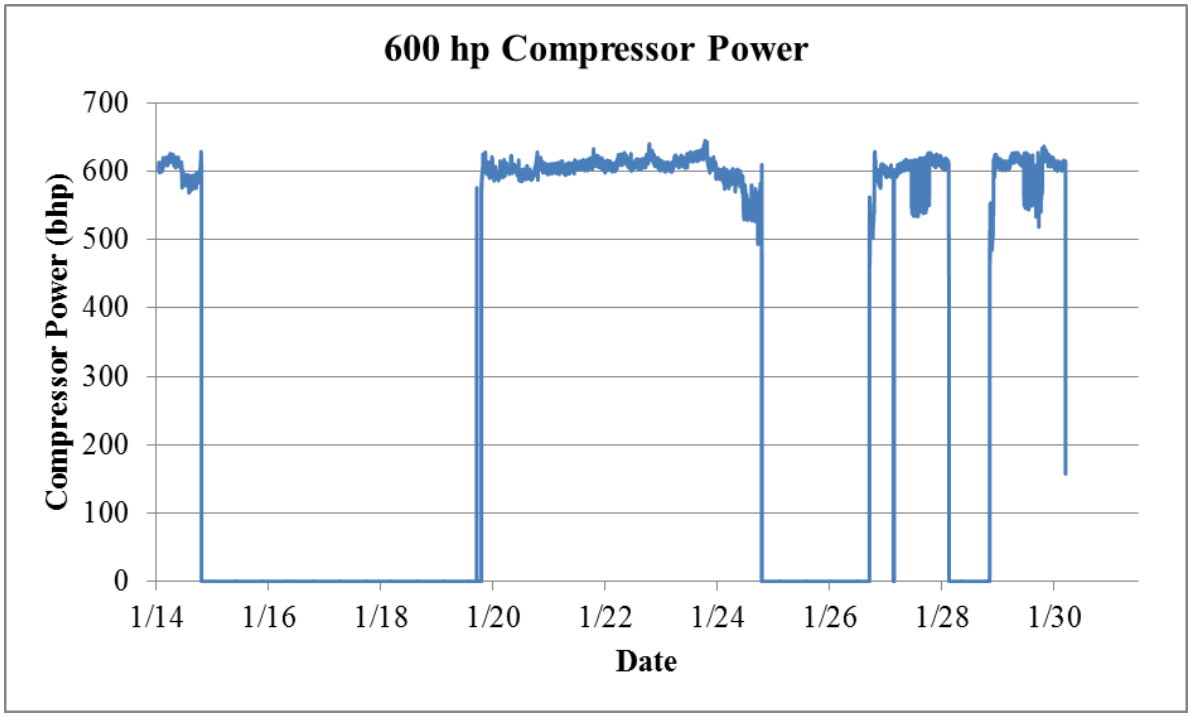


Figure 45: 600 hp Compressor Power

4.3.6 – Combined Analysis

To gain a better picture of how the system is operating as a whole, all of the air compressors power and system pressure is graphed in Figure 46 below. Pressure is indicated by the light blue curve on the top of the graph, followed by 600 hp compressor power trend in dark blue, the 500 hp compressor power trend in red, the 150 hp compressor power curve in green, and finally the 100 hp compressor power trend in purple.

As shown, the facility tends to operate multiple compressors simultaneously. During standard operation hours, which excludes weekends the facility tends to run either the 600 hp or the

500 hp compressors in conjunction with either the 150 hp compressor or the 100 hp compressor. On Saturdays, the plant typically operates the 150 hp compressor and the 100 hp compressor to fulfill compressed air needs, while on Sundays; the facility does not have a need for compressed air.

During the data collection period, the 500 hp compressor supplied compressed air to the facility from January 15th to January 18th. To fully meet the compressed air demand during this period, the 150 hp compressor was also turned on. From the 17th to the 18th of January, the 500 hp compressor power trended downwards, while the 150 hp compressor, utilizing modulating controls, maintained power. In the course of these operating conditions, the system pressure drifted upwards to 110 psig from 100 psig.

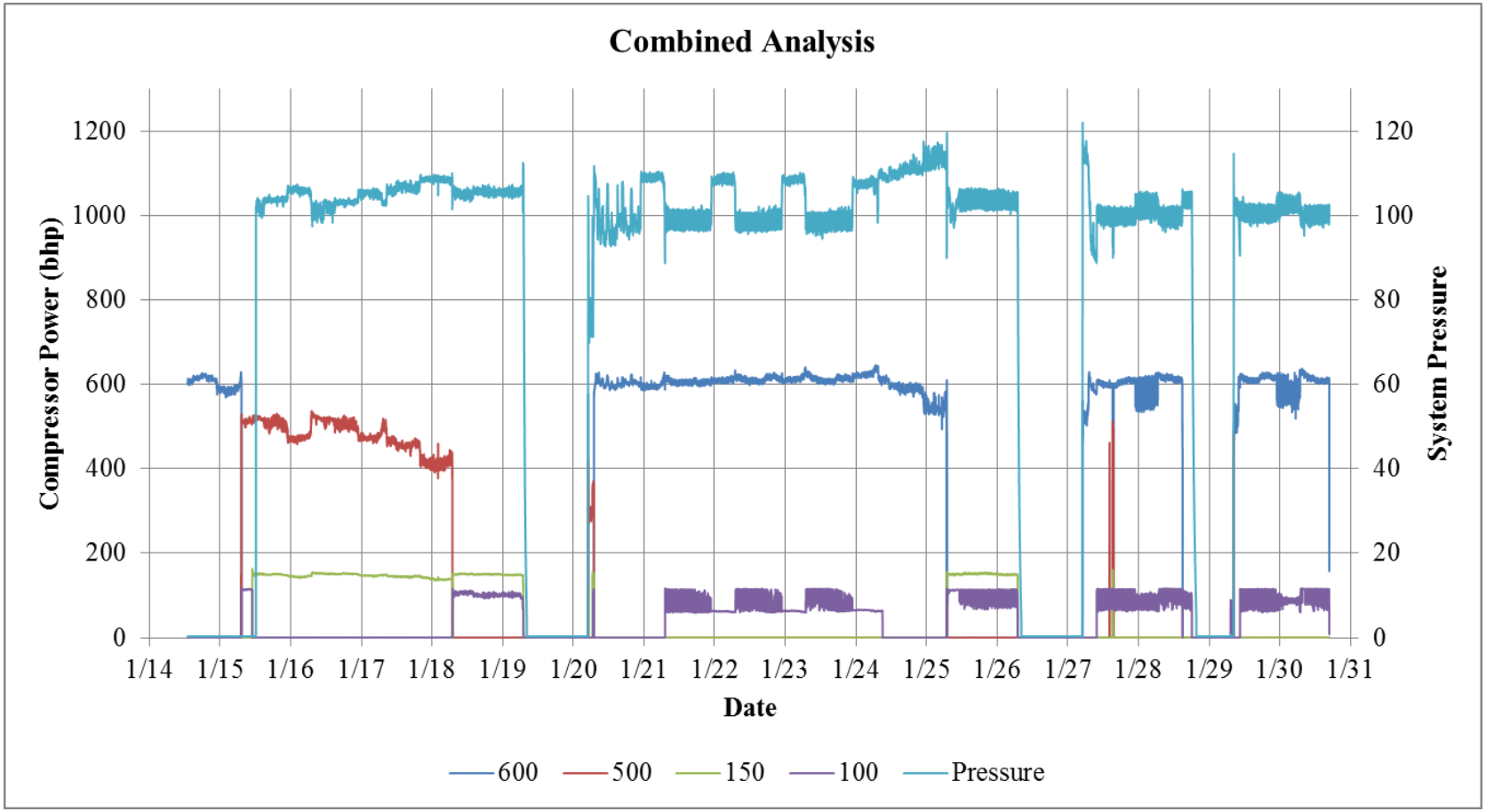


Figure 46: Plot of Compressor Power and Pressure

When the facility returns to operation after the weekend, on January 20th, all of the compressors turn on, but only the 600 hp compressor remains operating. From January 20th to the morning of January 21st, the 600 hp compressor is the only compressor running until the 100 hp compressor is used to trim for increased compressed air demand. From Figure 46, the 100 hp compressor is seen to operate in load/unload cycles during periods of increased demand, and remain unloaded during periods of low demand. The system pressure is expected to drop during high demand periods, and rise during low demand periods. This is clearly seen from the morning of January 21st to the morning of January 24th. Data from the morning of January 24th indicates that the 100 hp compressor shuts down, but the 600 hp compressor remains on until January 25th. The 600 hp compressor power trends lower from approximately 640 hp to 530 hp, until the morning of January 25th when it shuts down. During the period when the 600 hp compressor power trends lower, a pressure run up is observed in the data. The pressure rises from approximately 110 psig to 120 psig. This trend of lowering compressor power and increasing pressure indicates that the compressed air system is producing too much compressed air; implying less compressor power is required to satisfy the compressed air demand.

The final segment of data, January 27th to January 30th, shows the 600 hp air compressor and the 100 hp air compressor meeting the compressed air demand. It must be noted that during the compressor startup all of the compressors were initiated, but only the 600 hp and 100 hp compressors remained operating.

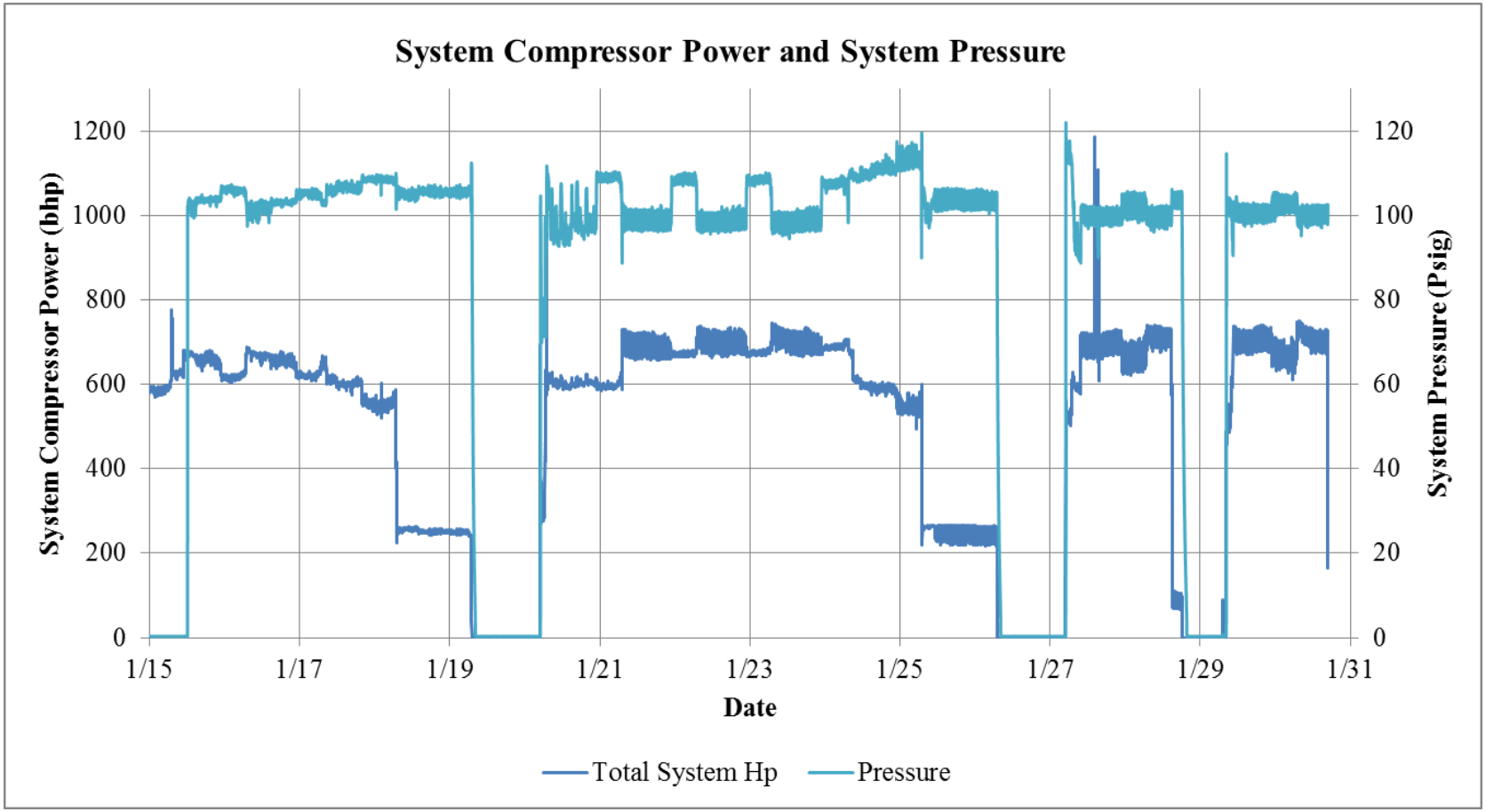


Figure 47: System Compressor Power and System Pressure

Figure 47 shows the total system compressor power and the corresponding system pressure. During the full production week, Monday through Friday, the average compressor power is 625 hp. On a typical Saturday the plant averages 337 hp of compressor power.

4.3.7 – Volumetric Flow Analysis

Due to the failure of the pressure data logger during the period of data collection from February 11th to February 27th, it was decided to use the amperage and pressure readings from the data collection period from January 15th to January 31st. The volumetric flow data from February 21st to February 27th will be utilized to predict to predict a volumetric flow for the data collection period From January 14th to January 30th. This will be accomplished by preforming a linear regression of a plot of volumetric flow against compressor power during the period from February 21st to February 27th. The linear regression will form a trend line with a corresponding equation for CFM per hp, which will be applied to total compressor power from January 14th to January 30th to estimate volumetric flow.

First, a brief description and analysis of the volumetric flow and system compressor power is needed. Volumetric flow data was recorded every 30 minutes, whereas the amperage recordings were every 5 seconds. To match volumetric flow readings with compressor power, the 5 second amperage readings were averaged over 30 minute periods. This means that 360 amperage data points were averaged together to create one 30 minute data point.

Figure 48 shows the compressed air flow in CFM from February 21st to February 28th. This data was measured and supplied by the facility. During the seven day period, the averaged compressed air demand was 2201 CFM. However, during production periods, the compressed air demand was 3075 CFM. The maximum compressed air demand was 3,783 CFM. Figure 49, is a plot of the system compressor power during this period. Figure 50 depicts both the compressed air volumetric flow and system compressor power on the same plot.

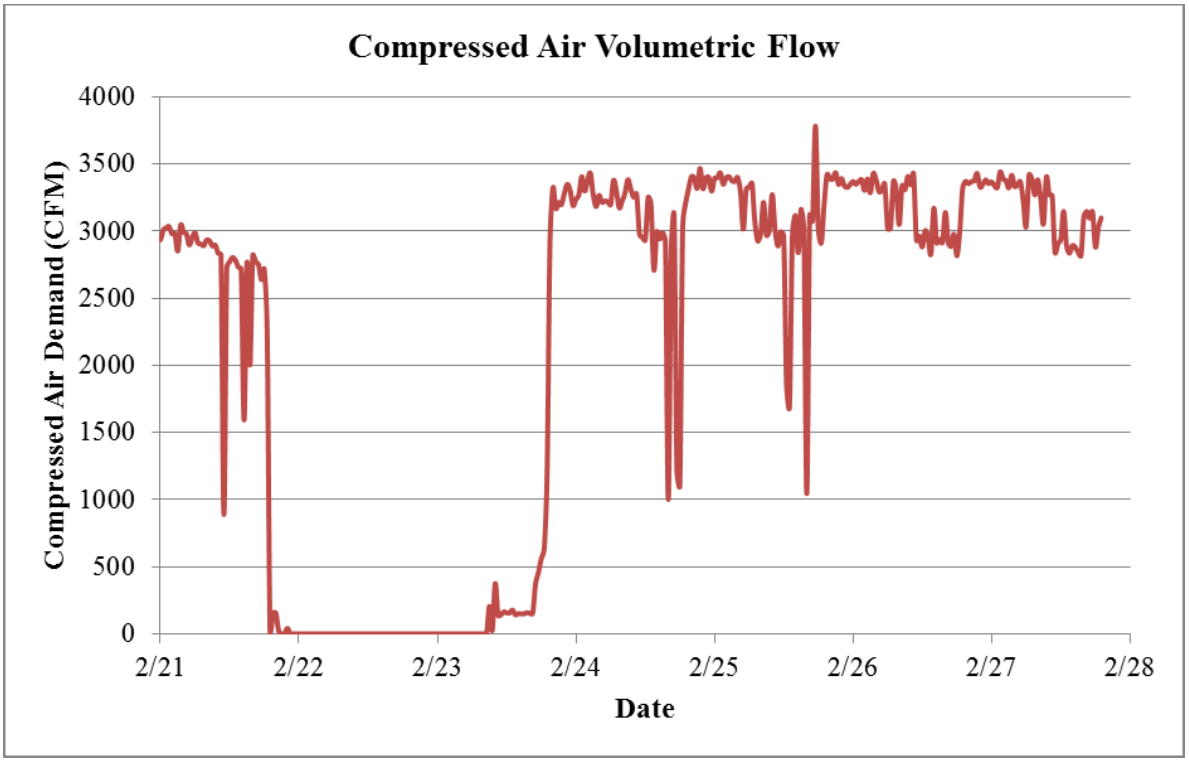


Figure 48: Compressed Air Demand (CFM)

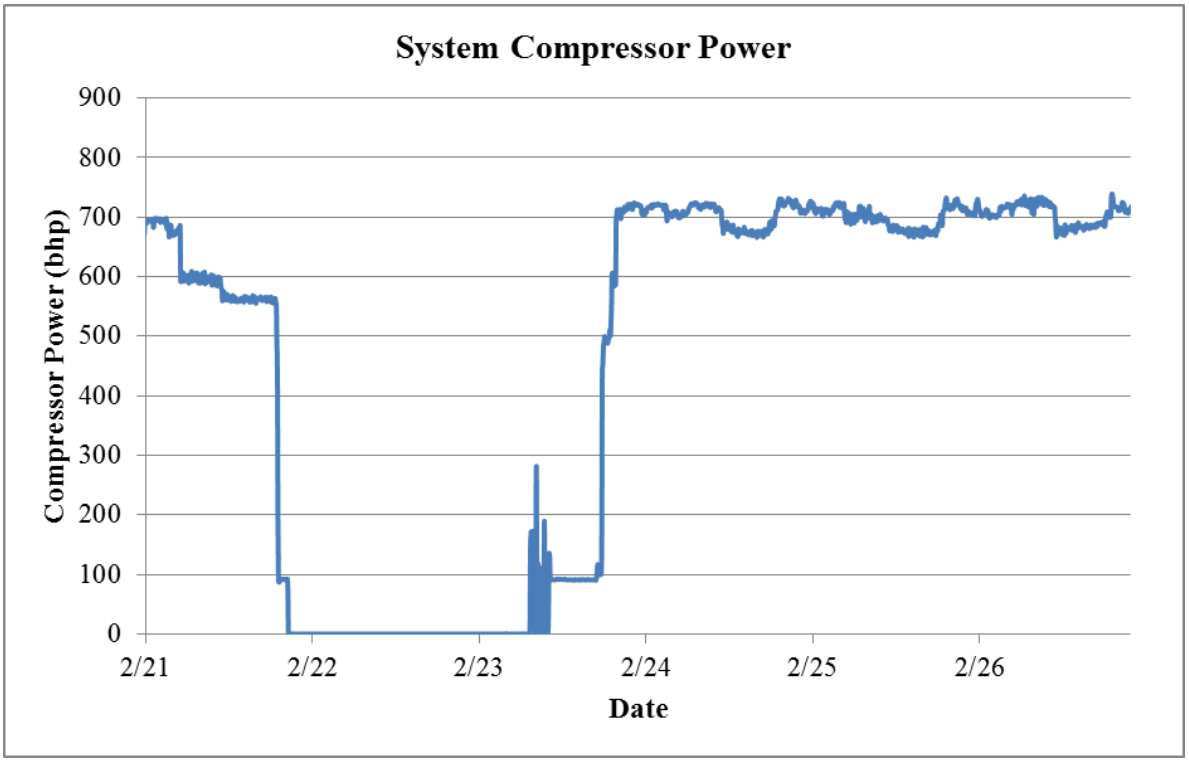


Figure 49: System Compressor Power February 21 to February 27

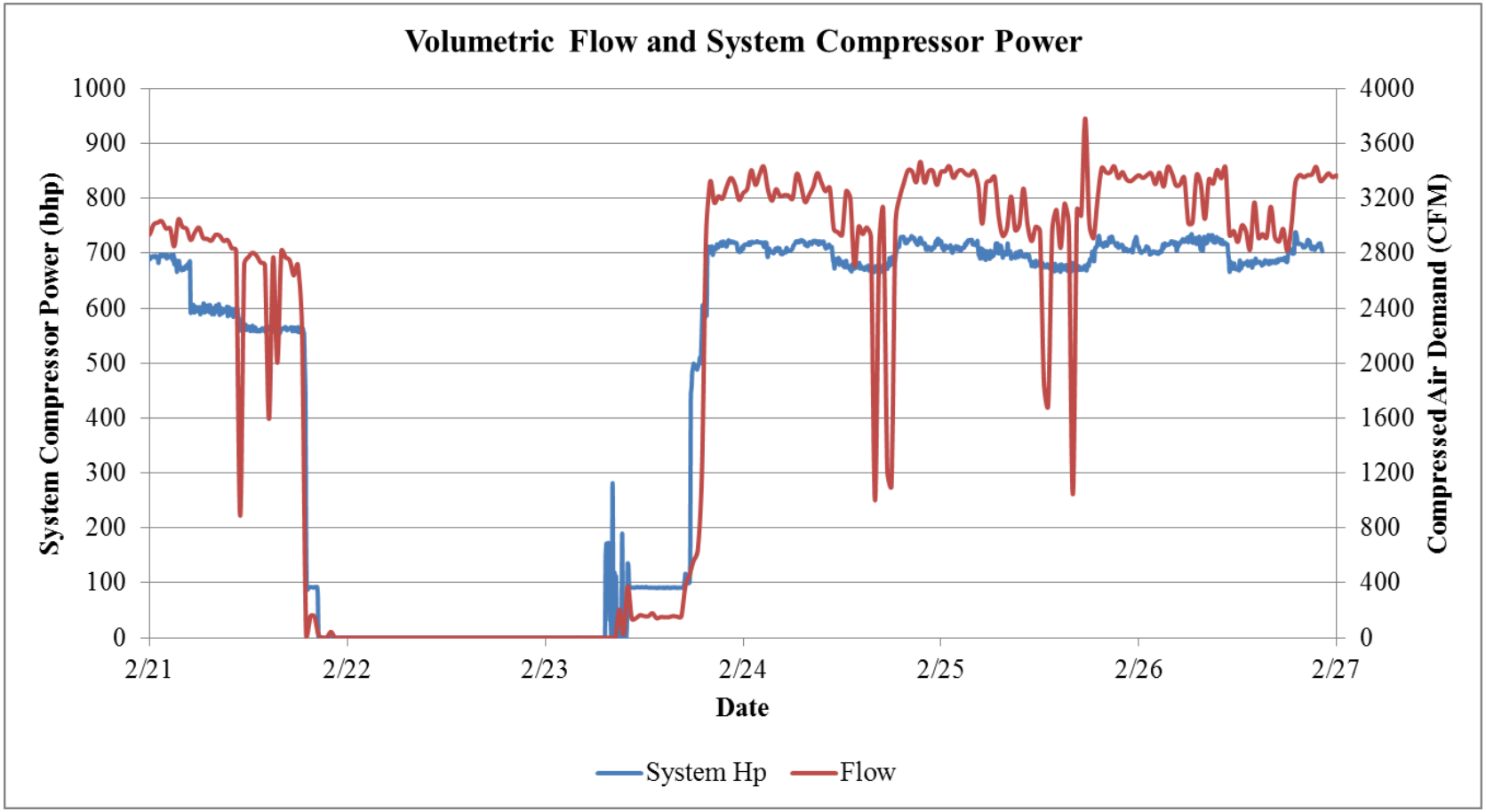


Figure 50: Compressed Air Demand and Compressor Power

Seen in Figure 50, the volumetric flow seems to correspond to the overall system compressor power. As the compressor power increases, the volumetric flow also increases. Similarly, as the compressors draw less power, the volumetric flow will also lower, and sometimes dramatically. The dramatic drops in compressed air flow, despite minor drops in the compressor power, is related to the individual compressor controls. The larger compressors utilize modulating controls, and therefore still use a considerable percentage of the compressor's power at low capacities.

To accurately estimate a volumetric flow, a curve fit was performed on a plot of volumetric flow against compressor power from February 21st to February 28th. A linear regression was performed, and the corresponding linear equation was added. This equation can be used to calculate the volumetric flow for periods without given values of compressed air demand. Figure 51 is a plot of the volumetric flow against system compressor power. The data indicates that there is a strong positive linear correlation between compressor power and the compressed air demand. Using the Pearson product-moment correlation equation, the correlation coefficient was found to be 0.968. The Pearson product-moment correlation coefficient is found with the following equation:

$$r = \frac{\sum\left(x-\bar{x}\right)\left(y-\bar{y}\right)}{\sqrt{\sum\left(x-\bar{x}\right)^2\sum\left(y-\bar{y}\right)^2}} \qquad \textbf{(Equation 5)}$$

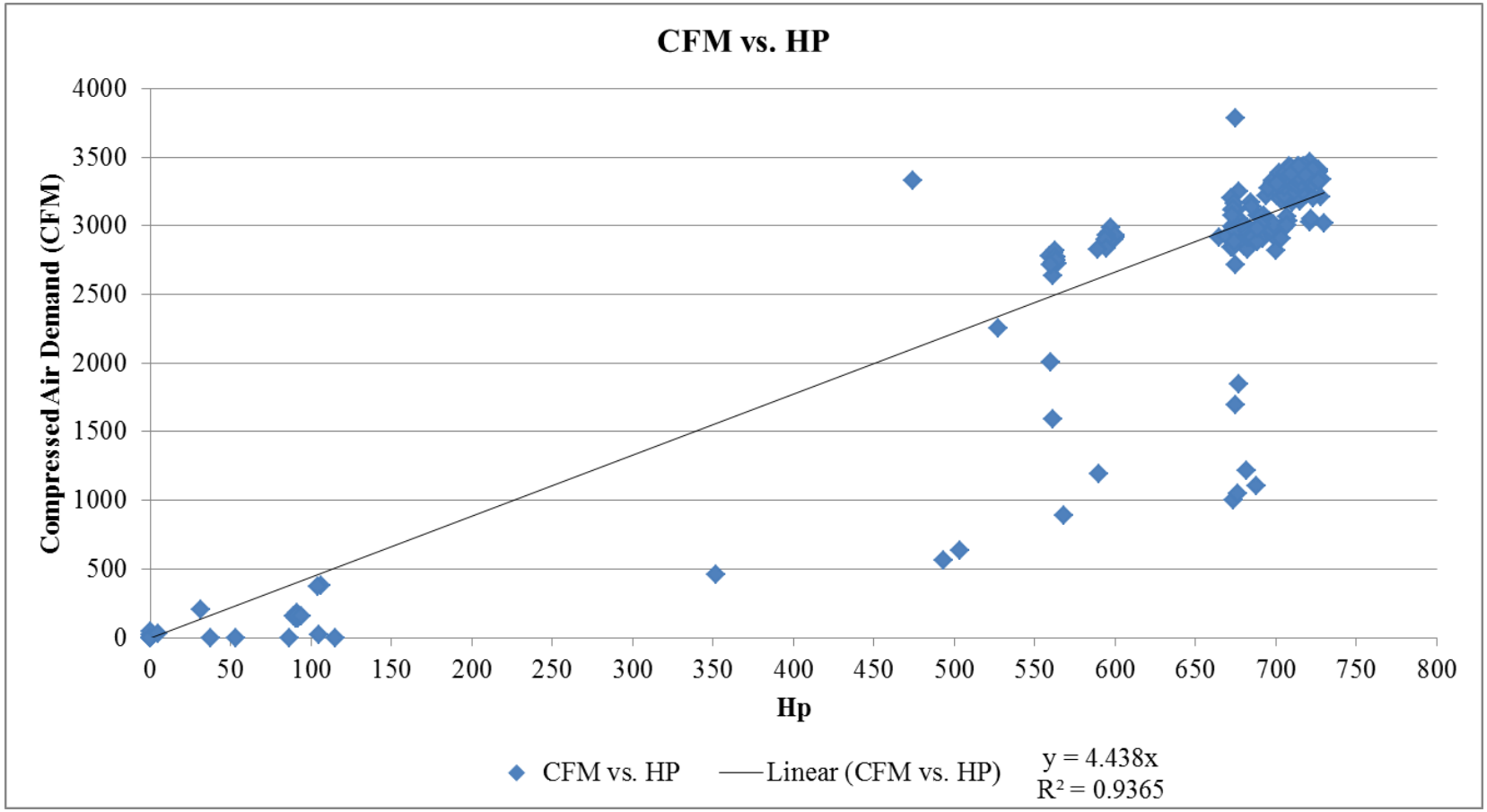


Figure 51: CFM vs. hp with Linear Regression

The linear regression results in a trend line formula of y=4.44(X). This means that the linear regression predicts that for every one system compressor horsepower 4.44 CFM will be produced by the air compressors. This is a reasonable estimate, as the CAGI reference for the 600 hp unit gives 4.4 CFM per horsepower at full package load. (13)

Figure 51 shows an interesting relationship between compressor power and compressed air volumetric flow rate. Generally, at high system compressor power, between 700 hp and 750 hp, the volumetric flow rate ranges from 2,700 CFM to 3,500 CFM. The modulating controls used for the 500 hp and 600 hp compressors help to explain the dips in volumetric flow at high compressor power, which is seen at around 650 hp and 700 hp. With modulating controls, the compressor requires a high percentage of its full rated power to produce a lower compressed air capacity. Additionally, we would expect lower compressed air flow at lower compressor power, which is clearly displayed in Figure 51.

The prediction of 4.44 CFM per horsepower will be more accurate at higher compressor capacities and less accurate when the larger compressors are modulating to a lower compressor capacity or during start up and shut down. This can be seen in Figure 52 below. The predicted volumetric flow in red generally trends with the actual volumetric flow, except for when the compressors are shutting down after production, turning on during production startup, or when the compressors modulate to a lower compressor capacity, but still use a high compressor power. To determine how similar the predicted volumetric flow is to the

actual volumetric flow, the percent difference was found for each point. The formula for percent difference is given by:

$$\%difference = \frac{|E1 - E2|}{\frac{1}{2} \times (E1 + E2)} \times 100 \quad \textbf{(Equation 6)}$$

The predicted volumetric flow during the shutdown on February 22nd had an average percent difference of 109%. Shutdown periods account for approximately 2.1 days per year. The startup on February 24th had an average percent difference of 101%. Startup periods account for approximately 4.4 days per year. During the times of low capacity, but high compressor power, the average percent difference is 57%. The system operates at low capacity, but high power approximately 2 days per year. However, the percent difference during periods of high capacity and high compressor power is only 3%.

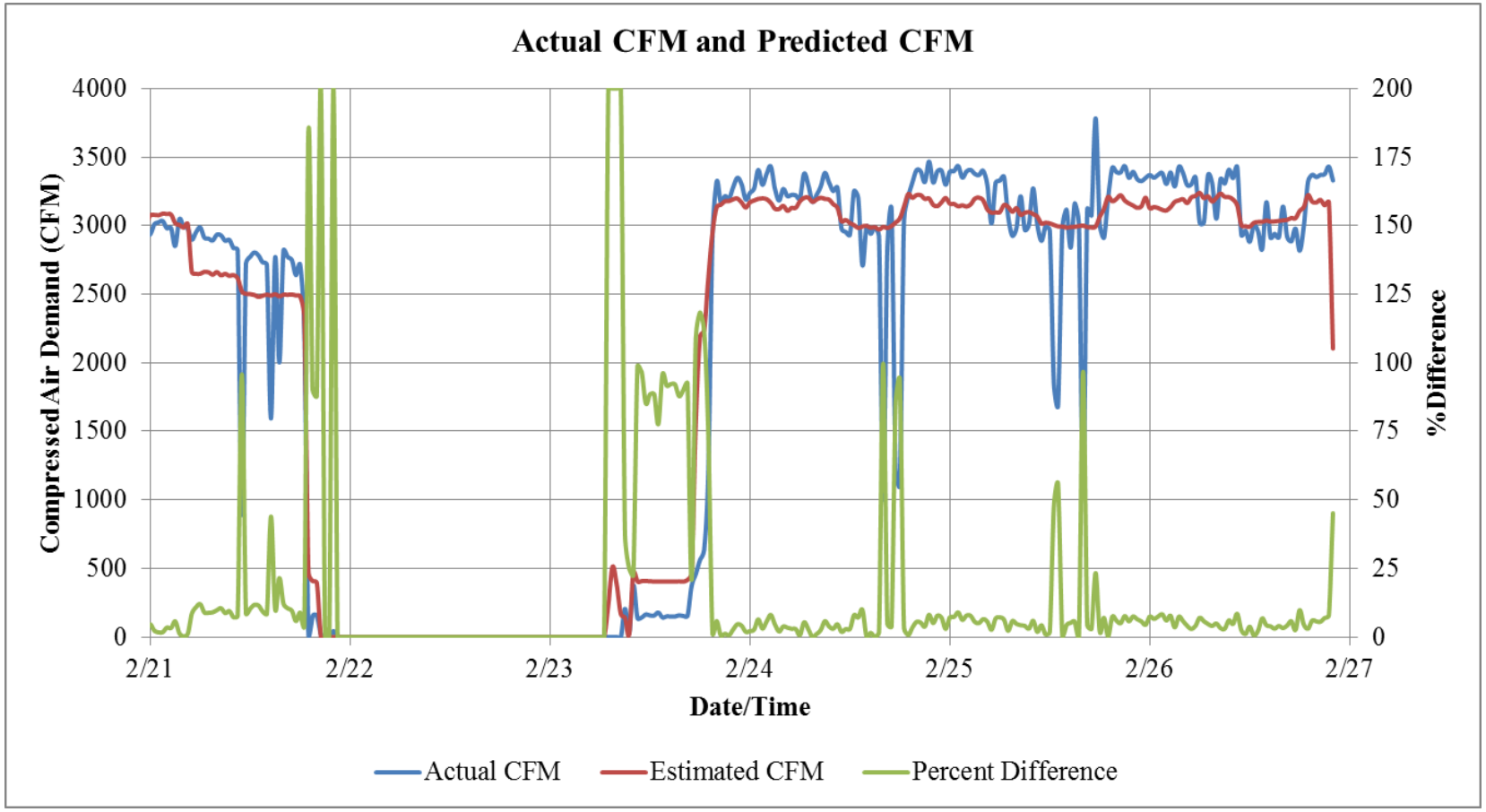


Figure 52: Actual CFM and Predicted CFM

Now, the volumetric flow will be estimated for the system compressor power data from January 14th to January 30th using the linear regression analysis previously performed. Although this will not accurately reflect the volumetric flow during shutdown, startup and low compressor capacity, it will adequately estimate the volumetric flow during normal compressed air demand periods.

Figure 53 is a plot of the predicted volumetric flow from January 14th to January 30th. The plot shows large spikes in volumetric flow when all of the compressors turn on during production startup or during increased demand. The volumetric flow estimates during this time are likely inaccurate because of the time it requires for a compressor to actually start producing compressed air. The system almost immediately turns the redundant compressors off before they can start to produce compressed air.

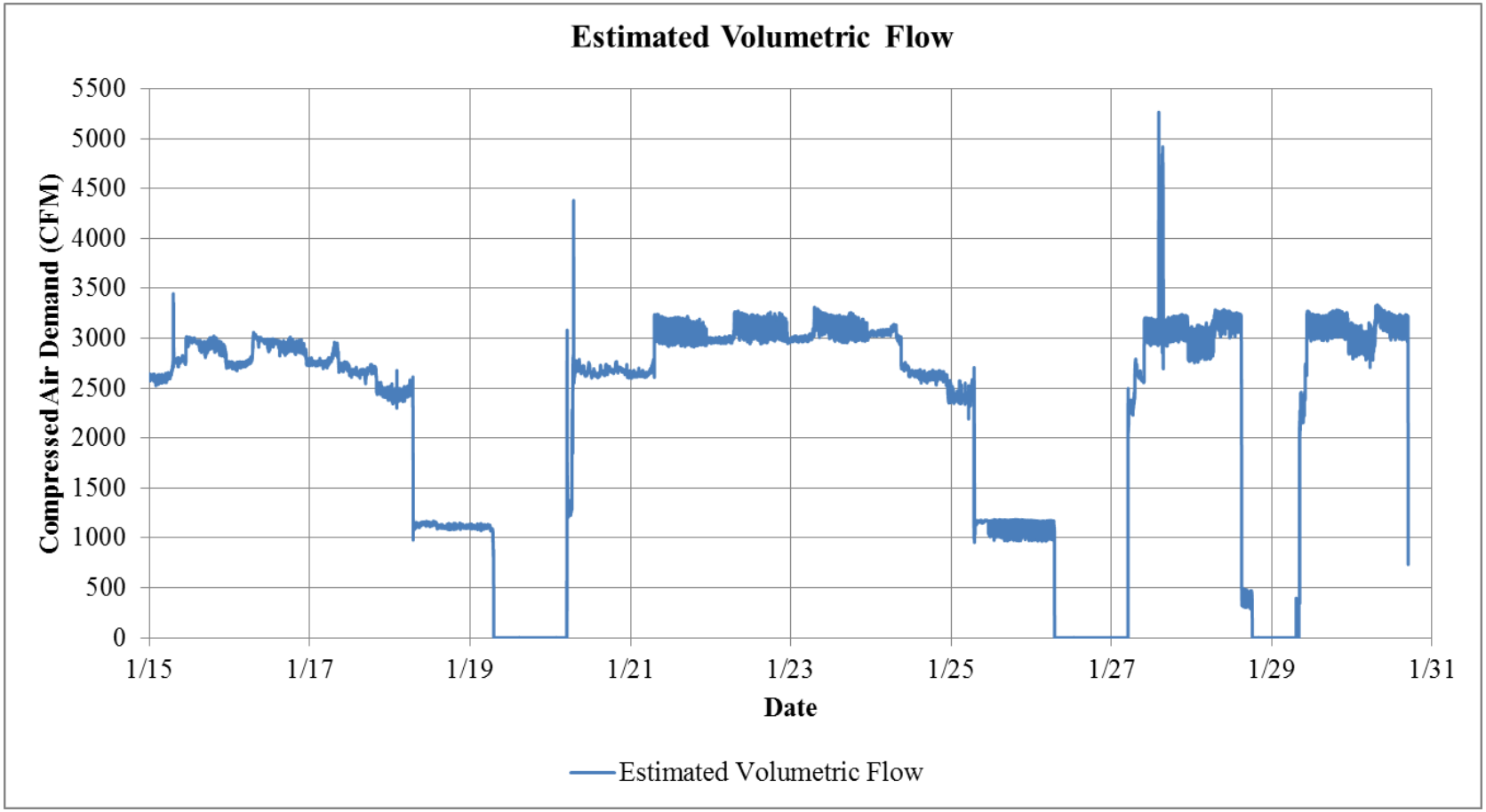


Figure 53: Estimated Volumetric Flow

Because the large spikes in the estimated volumetric flow plot are unlikely, and similar spikes in actual volumetric flow are not seen, these points will be removed. These changes are reflected in Figure 54 on page 93.

Figure 55 on page 94 is a plot comparing the system pressure in psig to the estimated volumetric flow. Pressure and volumetric flow are depicted in red and blue respectively. Generally, as the volumetric flow in a compressed air system increases the system pressure will reduce, indicating a demand event. This is seen from January 20th to January 24th, as the pressure drops when the compressed air demand increases. This trend is normal and expected, but another trend emerges in the data, which is concerning. From January 17th to January 18th the pressure rises while the volumetric flow decreases, which is known as pressure run up. This occurs during periods of low demand, meaning that compressor power and capacity are decreasing while the pressure continues to increase. Ideally the pressure would remain at a set point of around 95 psig. Operating the compressors at 120 psig uses an unnecessary amount of compressor power, and wastes money by increasing the amount of energy consumed by the air compressor. This operating trend occurs again from January 24th to January 25th.

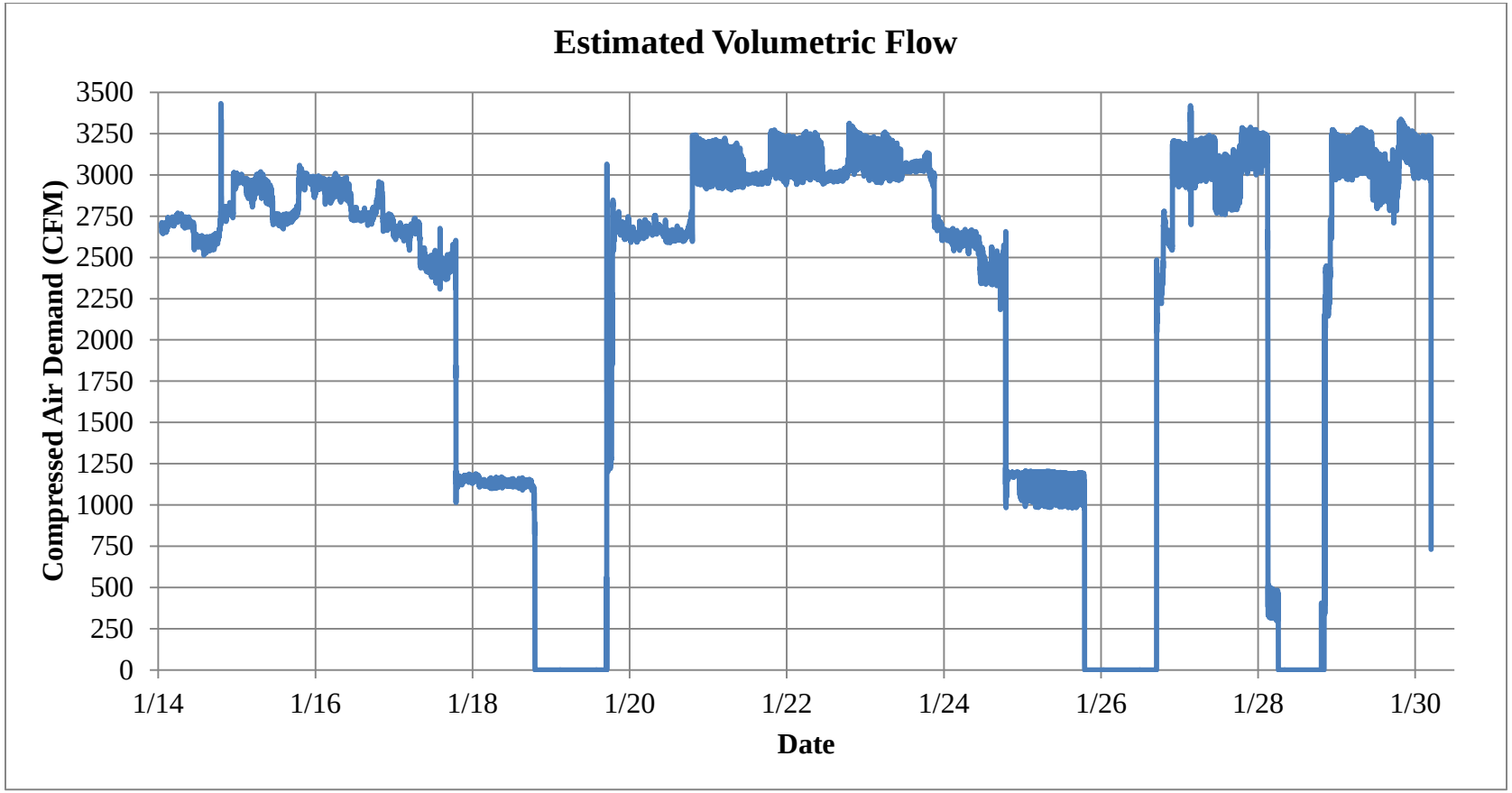


Figure 54: Estimated Volumetric Flow Excluding Outliers

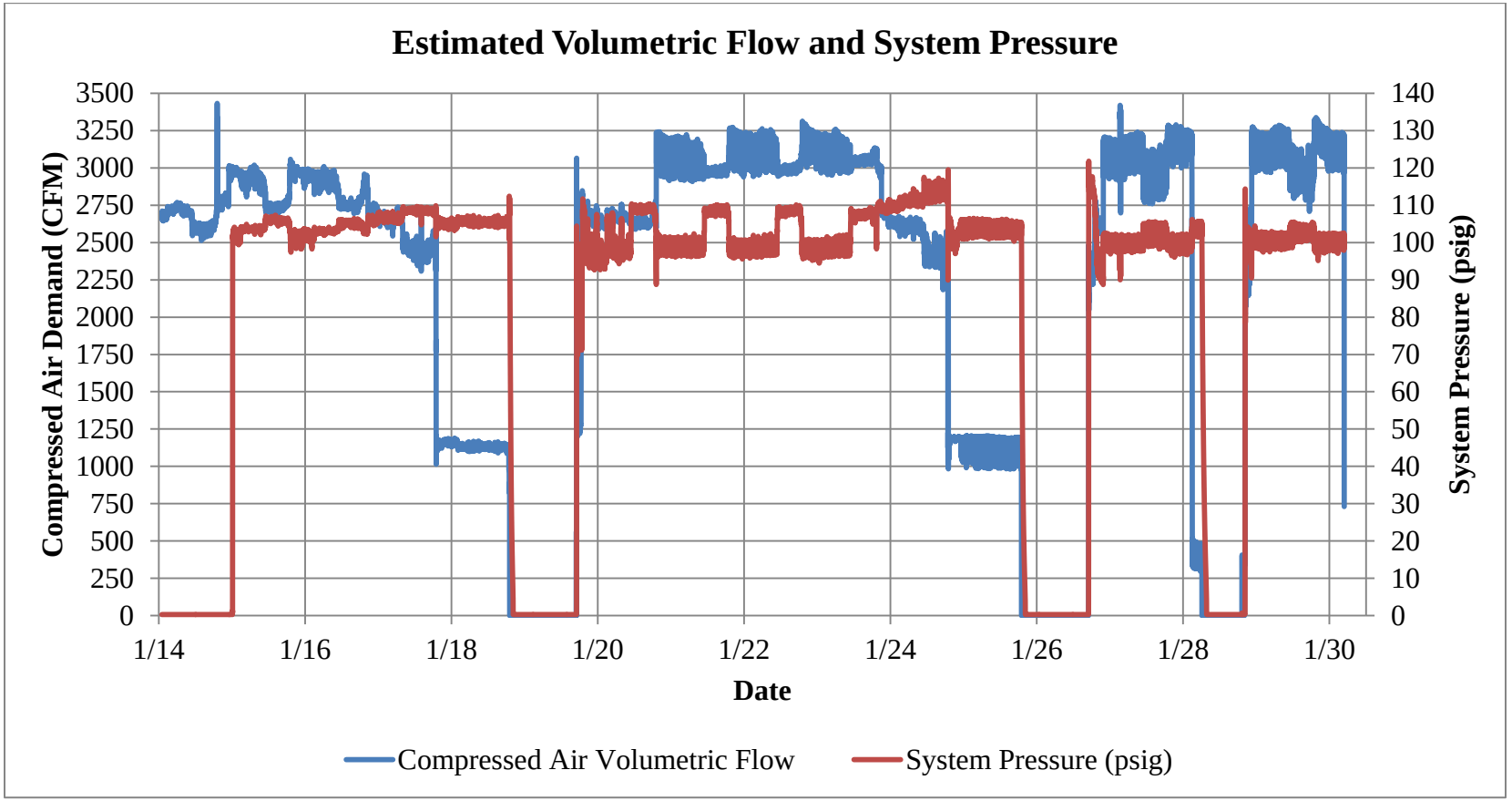


Figure 55: Estimated Volumetric Flow and System Pressure

4.3.8 – Pressure Analysis

The pressure varies greatly depending on the system compressor power and compressed air demand. Figure 56 is a plot of pressure in psig against the combined compressor power.

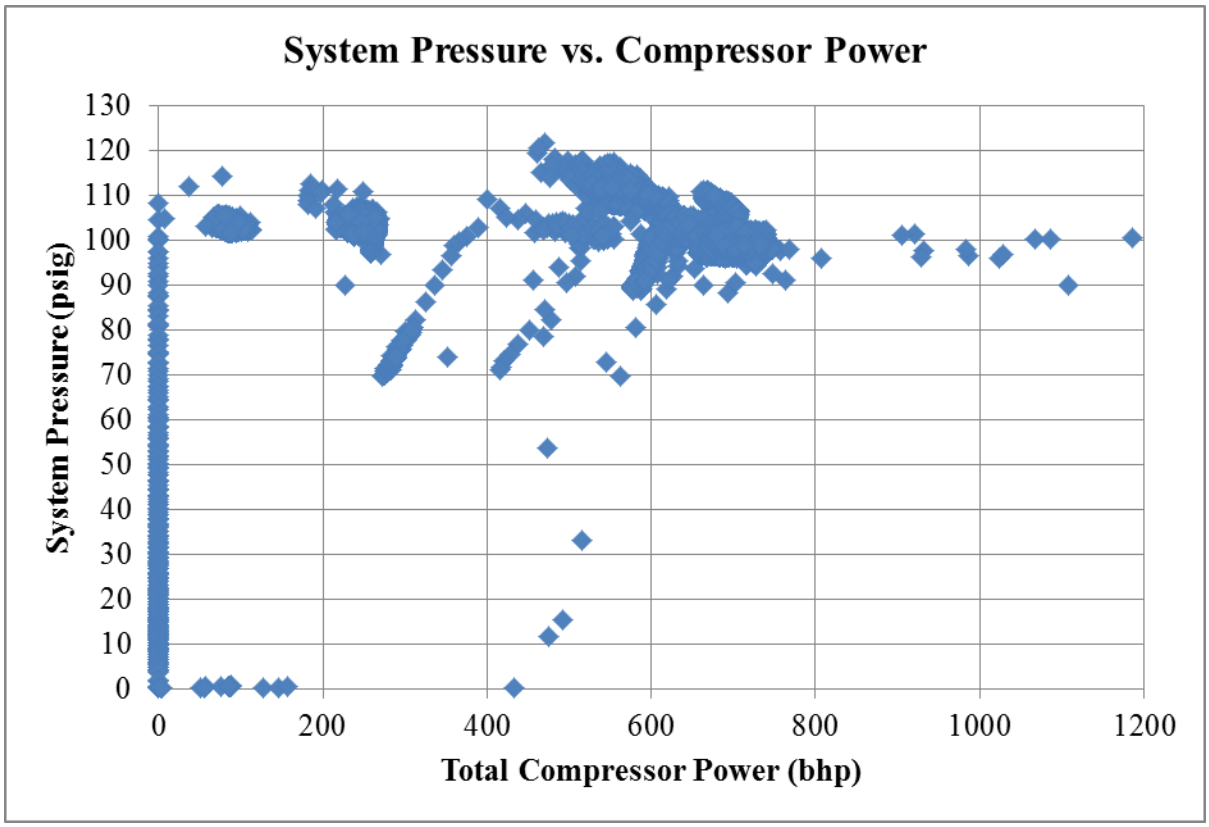


Figure 56: System Pressure Varying with Compressor Power

There are some interesting observations regarding Figure 56. At zero hp the pressure ranges from 110 psig to 0 psig. This is due to pressure bleed down after the compressors are shut down for Sundays. The next grouping of data points in Figure 56 shows the system pressure

when the 100 hp compressor is the only compressor running, which is between 70 hp and 130 hp on Figure 56. The power ranges from about 71 hp to 125 hp. The pressure reaches about 105 psig, and the minimum is roughly 100 psig. It appears that during high demand periods the pressure remains between 100 psig and 105 psig, but during low demand periods the will rise above 105 psig. This is seen in Figure 57 below. The data points reading zero psig corresponding to positive power likely is when the compressor first turns on.

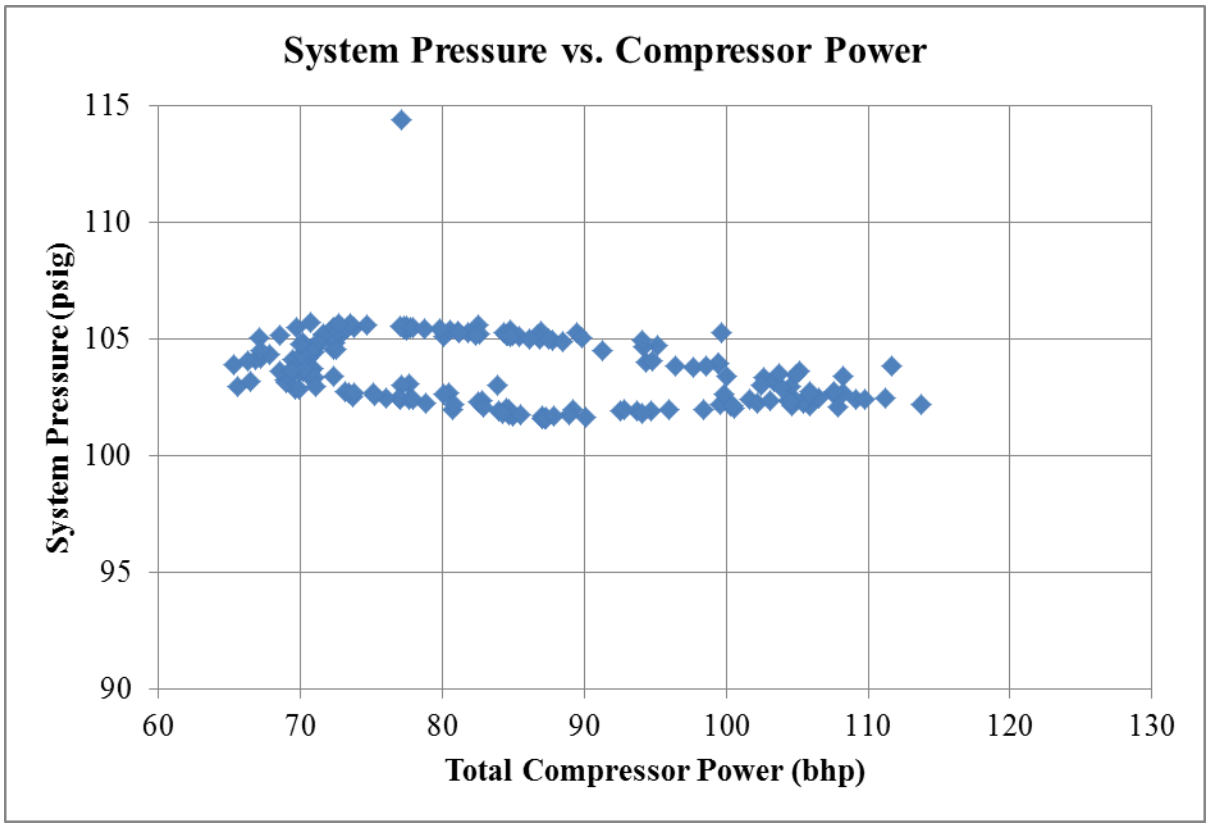


Figure 57: Pressure vs. Power (60 hp-120 hp)

The third grouping of data points, between 210 hp and 370 hp in Figure 56, is likely the system pressure when the combination of the 150 hp compressor and the 100 hp compressor are operating. A similar pattern to the pressure vs. 100 hp compressor occurs in when this combination of compressors are running. This operating condition is regularly seen on Saturdays, when the demand for compressed air is lower. The lowest pressure recorded while the compressors were producing compressed air is 97 psig at 258 hp, while the maximum is 111 psig at 249 hp. During low loads the pressure remains between 103 psig and 105 psig. The pressure range is higher during high loads, typically remaining between 100 and 105 psig. Pressure against compressor power is shown in Figure 58.

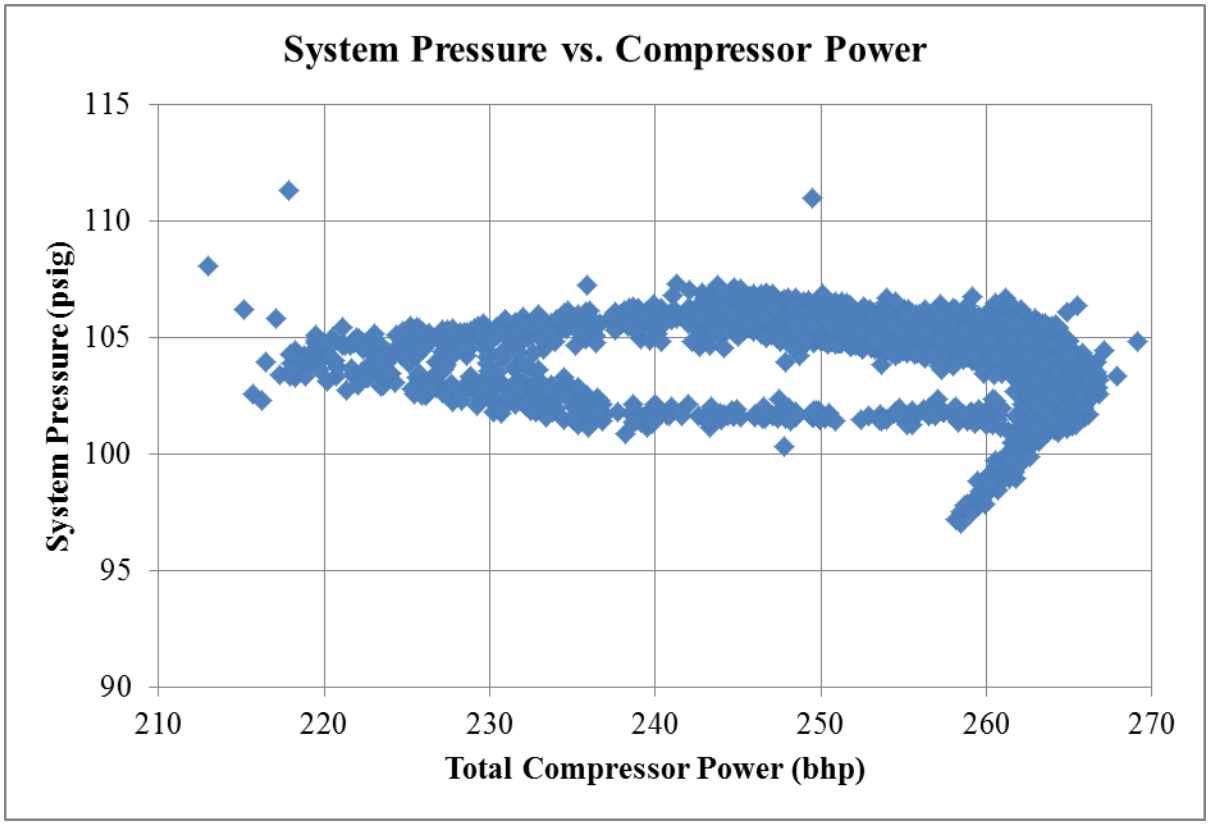


Figure 58: System Pressure vs. Compressor Power (210 hp to 270 hp)

The final grouping of data points from 325 hp to 825 hp shows a few different combinations of compressors. It includes the 600 hp compressor operating alone, the 600 hp compressor operating with the 100 hp compressor, and the 500 hp compressor operating with the 150 hp compressor. The most notable trend occurs between 500 hp and 800 hp.

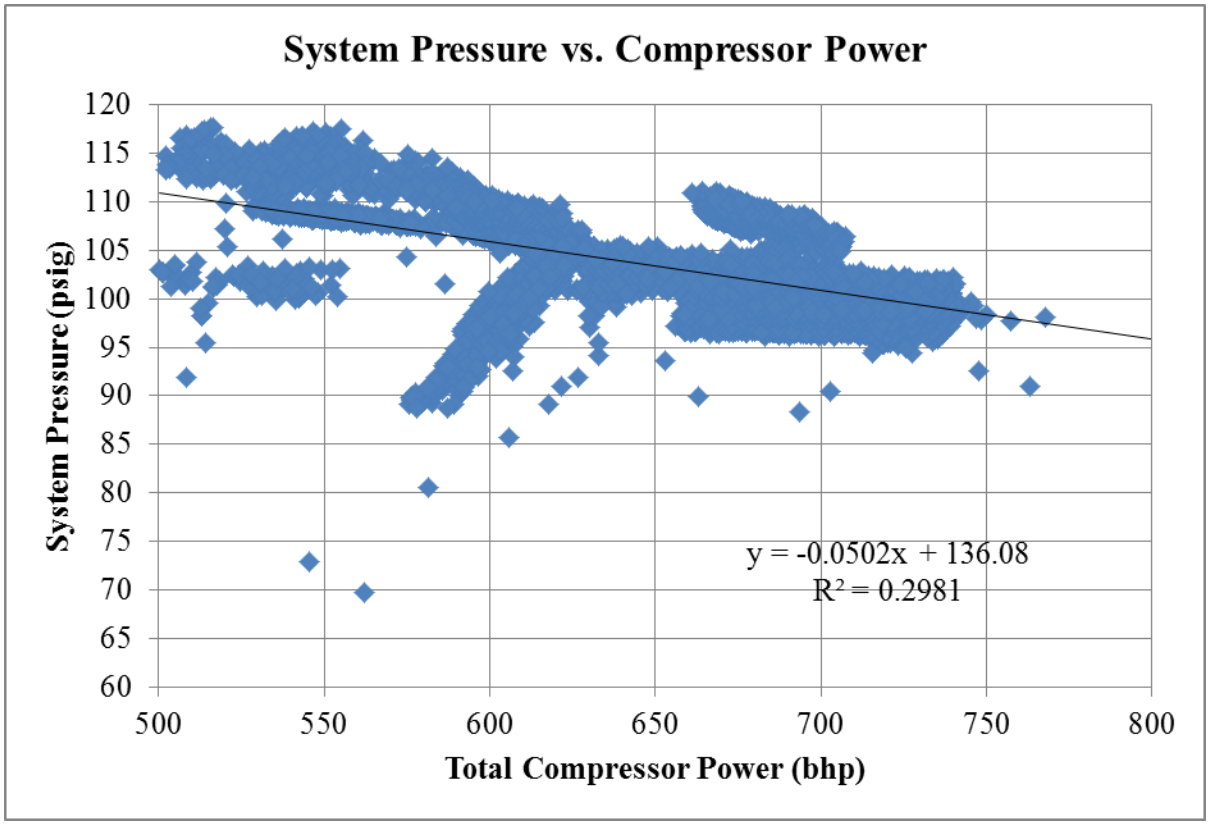


Figure 59: Pressure vs. Compressor Power (500 hp to 800 hp)

Figure 59 shows pressure decreasing as the compressor power increases. This is a typical trend seen with modulating compressors as compressed air demand increases. At low demand, pressure is high and the compressors can operate at a lower power percentage. As the compressed air demand increases, the butterfly valve begins to open up and the compressor power increases. The pressure continues to decrease until the compressors are at full capacity and the demand is met. Figure 59 indicates a negative correlation between system pressure and total compressor power at higher percentages of compressed air system power.

Chapter 5 - AIRMaster+ System Modeling

AIRMaster+ is an energy efficiency tool used to model compressed air systems, and then create energy efficiency recommendations. It can help to maximize compressed air system efficiency.

AIRMaster+ allows the user to manage multiple facilities and compressed air systems, cataloging an inventory of generic or name brand air compressors. The software simulates the existing compressed air system and any modifications. The software will model part load system operation for any combination of compressors. AIRMaster+ will allow the modeler to consider a number of efficiency measures including; reducing air system air leaks, improving end use efficiency, reducing the compressor air pressure, adjusting cascading set points, implement unloading controls when applicable, implement an automatic sequencer, programming compressors to run less or more depending on compressed air demand, and finally add receiver volume if necessary (14).

5.1 Modeling the Compressed Air System

Modeling a facility's compressed air system involves entering factual information about the company, utilities, facility, compressed air system operation, the compressors, and creating a profile based on data acquired by data logging. After opening the AIRMaster+ program, the

user is presented with buttons to input, review, and edit the required information. The figure below depicts the home screen of the program.

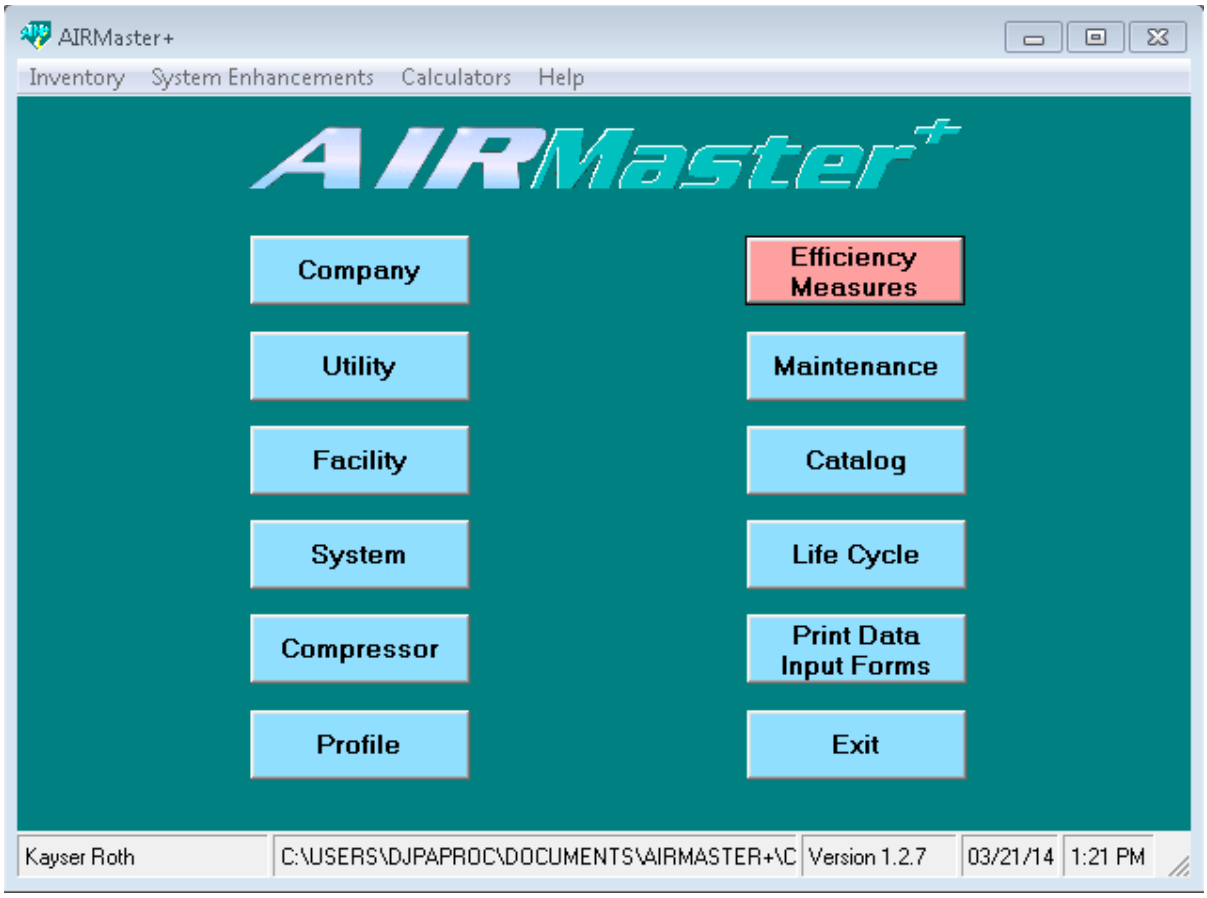


Figure 60: AIRMaster+ Home Screen (15)

5.1.1 Company Module

After clicking on the company button, the user is presented with several inputs. After indicating the name of the company, the industry type, SIC description, the units, and the currency type, the user can save and move forward to the next step.

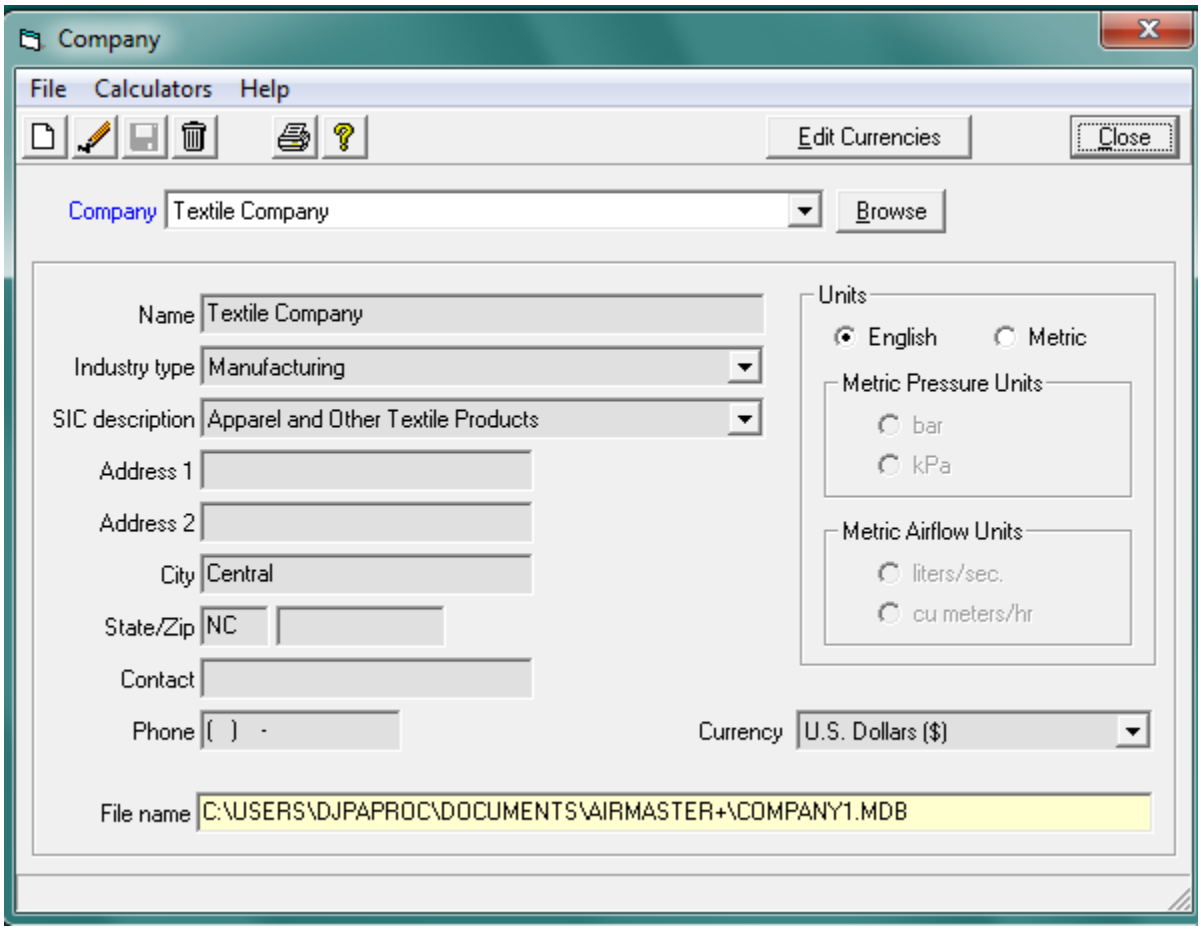


Figure 61: AIRMaster+ Company Screen (15)

5.1.2 Utility Module

Determining the electricity rate is an important step in calculating cost savings from energy efficiency recommendations. For this step, the average energy and demand rates were used as inputs. The electricity rates are divided into two seasons, but as the electricity rates were averaged over a 12 month period, both seasons are equivalent. Energy costs $0.0429/kWh and demand costs $11.63/kW.

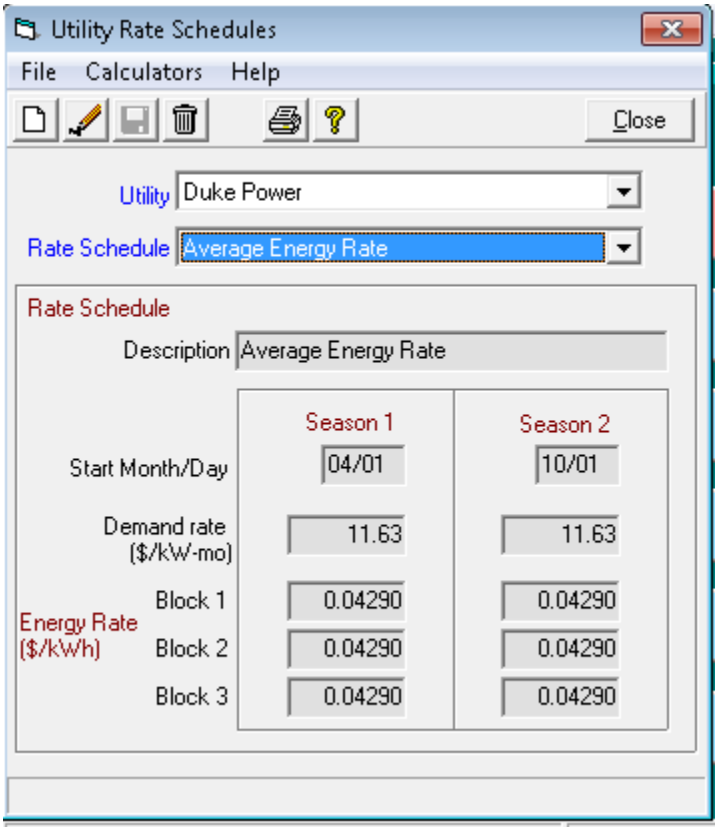


Figure 62: AIRMaster+ Utility

5.1.3 Facility Module

There are two main tabs within the Facility setup; Facility Information, and Compressor Summary (from Compressor module). The Facility Information tab includes input for the electricity rates, facility name, and annual plant electrical use in kWh. The utility information fills in automatically when the facility name is chosen from the drop down menu on the top left corner of the screen. The second tab, Compressor Summary is automatically filled in after the compressor information is recorded in the Compressor module.

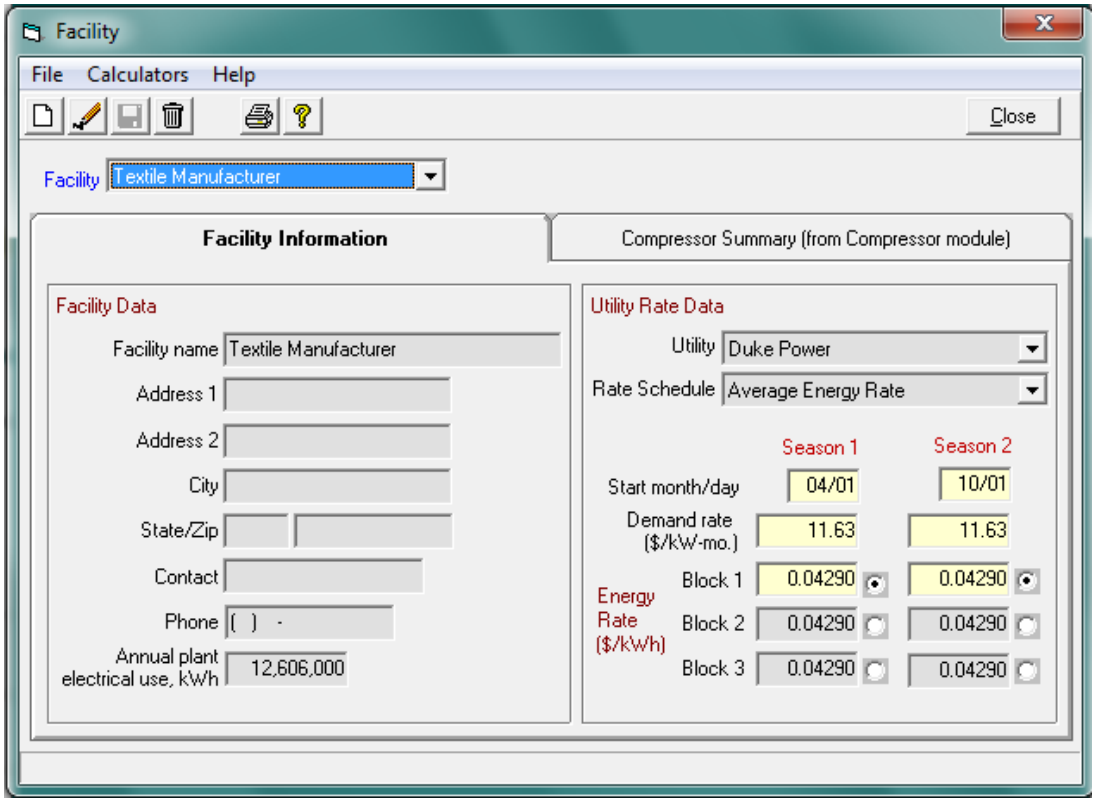


Figure 63: AIRMaster+ Facility Information (15)

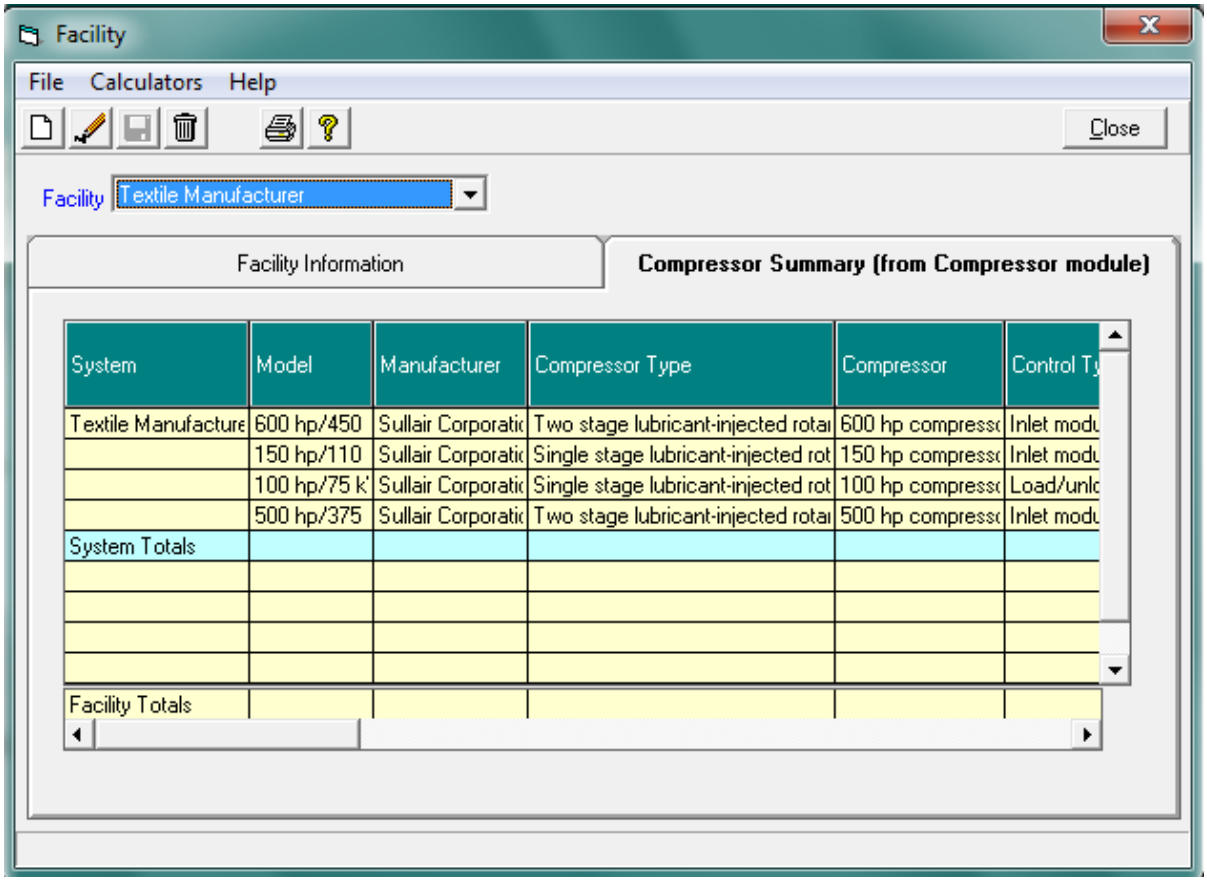

Facility
File Calculators Help
Close
Facility Textile Manufacturer
Facility Information
Compressor Summary (from Compressor module)
System Model Manufacturer Compressor Type Compressor Control Ty
Textile Manufacture 600 hp/450 Sullair Corporatic Two stage lubricant-injected rotai 600 hp compressc Inlet modu
150 hp/110 Sullair Corporatic Single stage lubricant-injected rot 150 hp compressc Inlet modu
100 hp/75 k' Sullair Corporatic Single stage lubricant-injected rot 100 hp compressc Load/unlc
500 hp/375 Sullair Corporatic Two stage lubricant-injected rotai 500 hp compressc Inlet modu
System Totals
Facility Totals

Figure 64: AIRMaster+ Facility Compressor Summary (15)

5.1.4 System Module

The System Data tab, within the System module allows the user to input the nominal system pressure, elevation of facility, and air storage capacity. The user can also enter sequencer data, should the system utilizes a sequencer.

The Daytypes tab allows the user to define the types of days, in which the compressors operate. Each daytype is assigned a number of operating days. Examples of daytypes could

be production days, or weekends. For this project, Production, Saturdays, and Sundays were used as daytypes. These daytypes are used later to input averaged logger data for specific periods of operation.

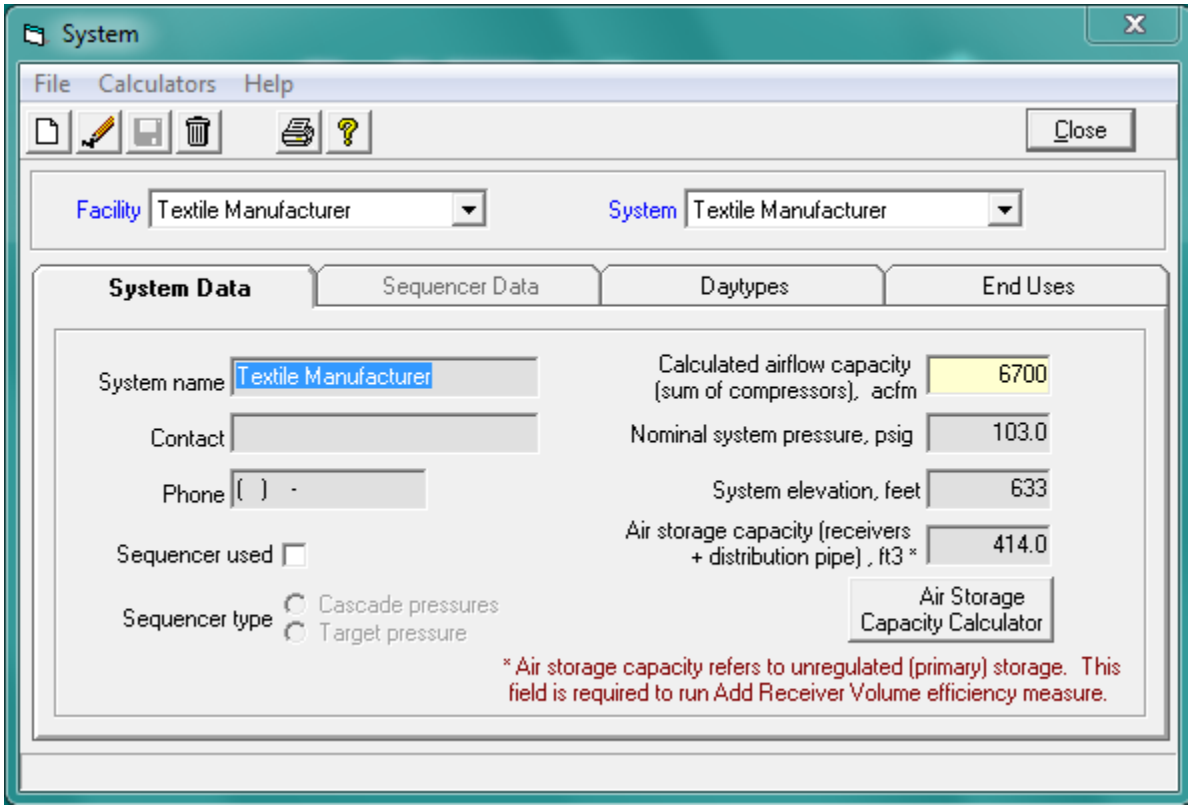


Figure 65: AIRMaster+ System Module (15)

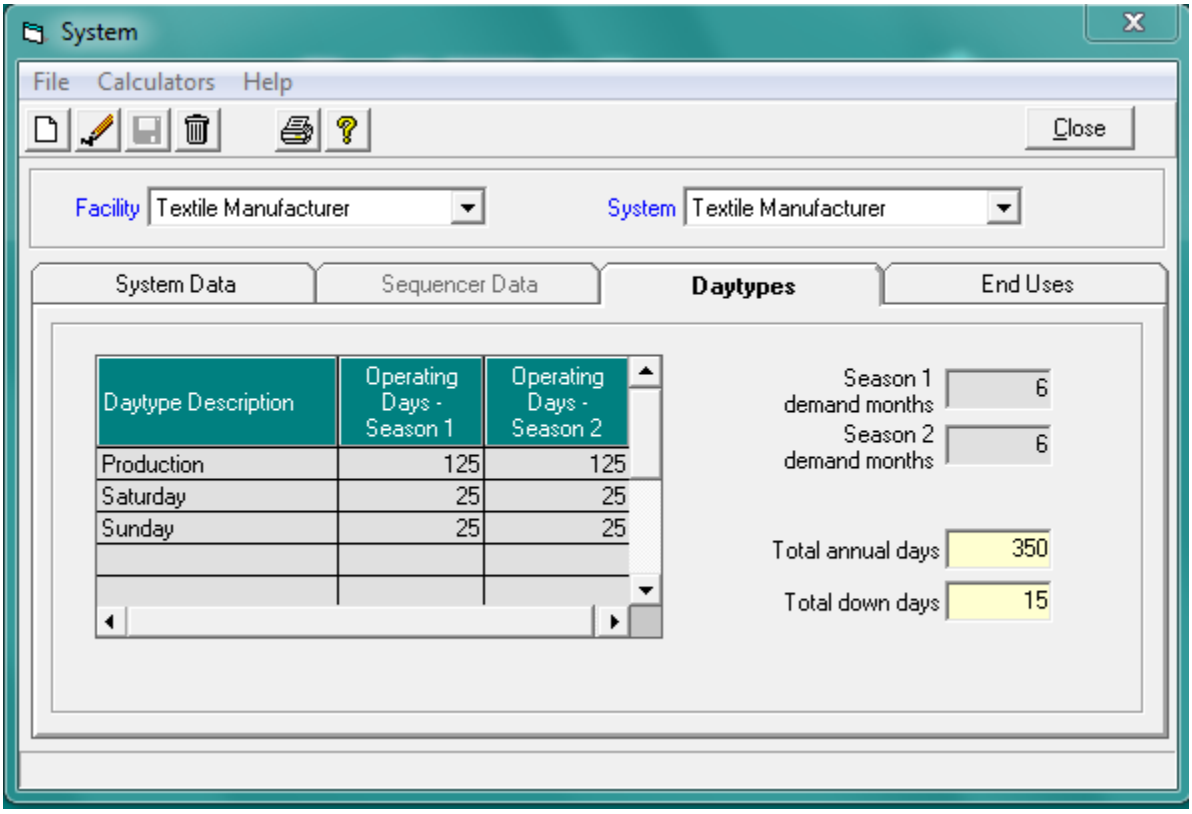


Figure 66: AIRMaster+ System Module Daytypes (15)

The System module also allows the user to qualify compressed air end uses; however this project is not concerned with end use.

5.1.5 Compressor Module

The Compressor module is used to add compressors to the system. Once the compressors are added to the system, the controls and the performance can be modified for each compressor.

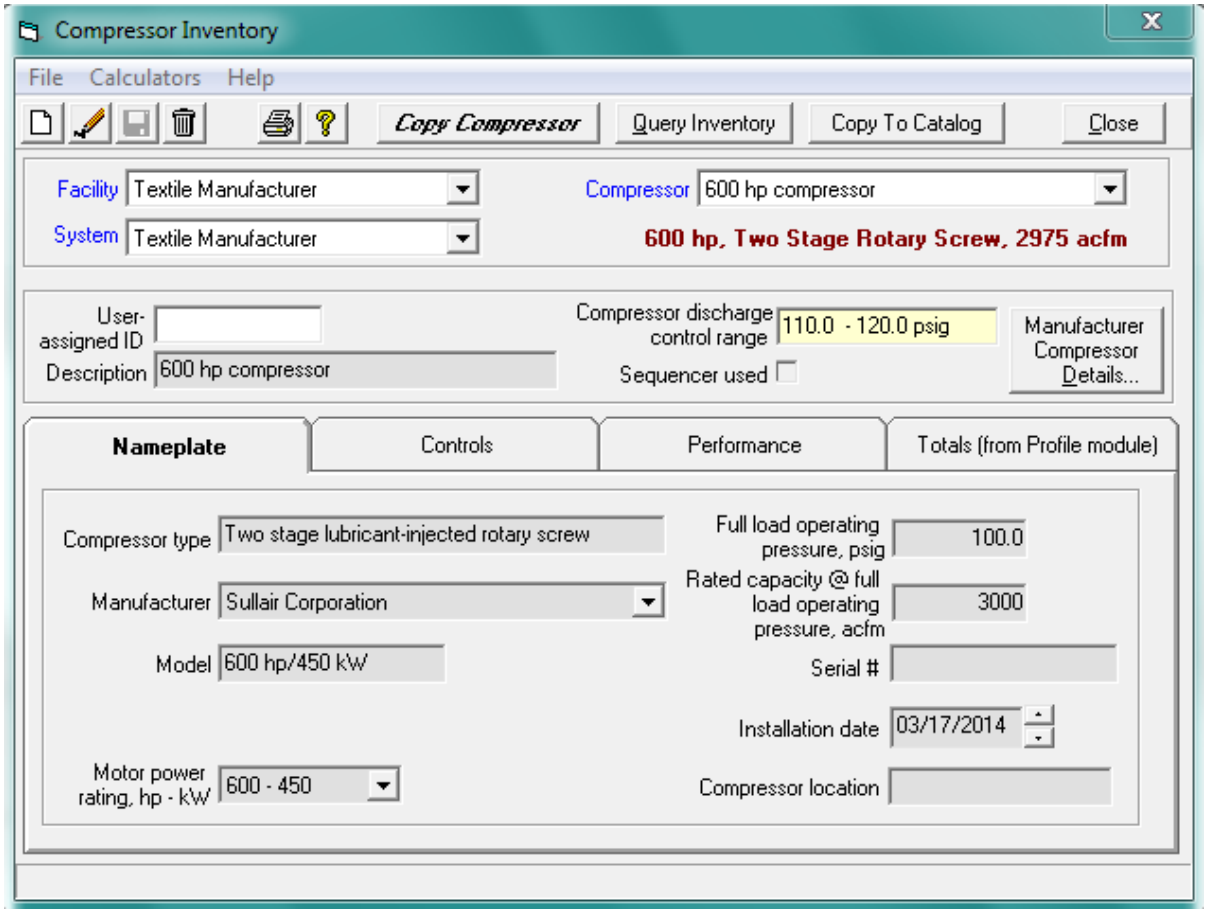


Figure 67: AIRMaster+ Compressor Module (15)

The first step is to add a new compressor to the system, which is accomplished by clicking *Add New Record* on the top left corner of the screen, under the File tab. This will bring up a screen, in which the user can search for a compressor by compressor type, control type, or power rating. The compressor catalog is shown in Figure 68.

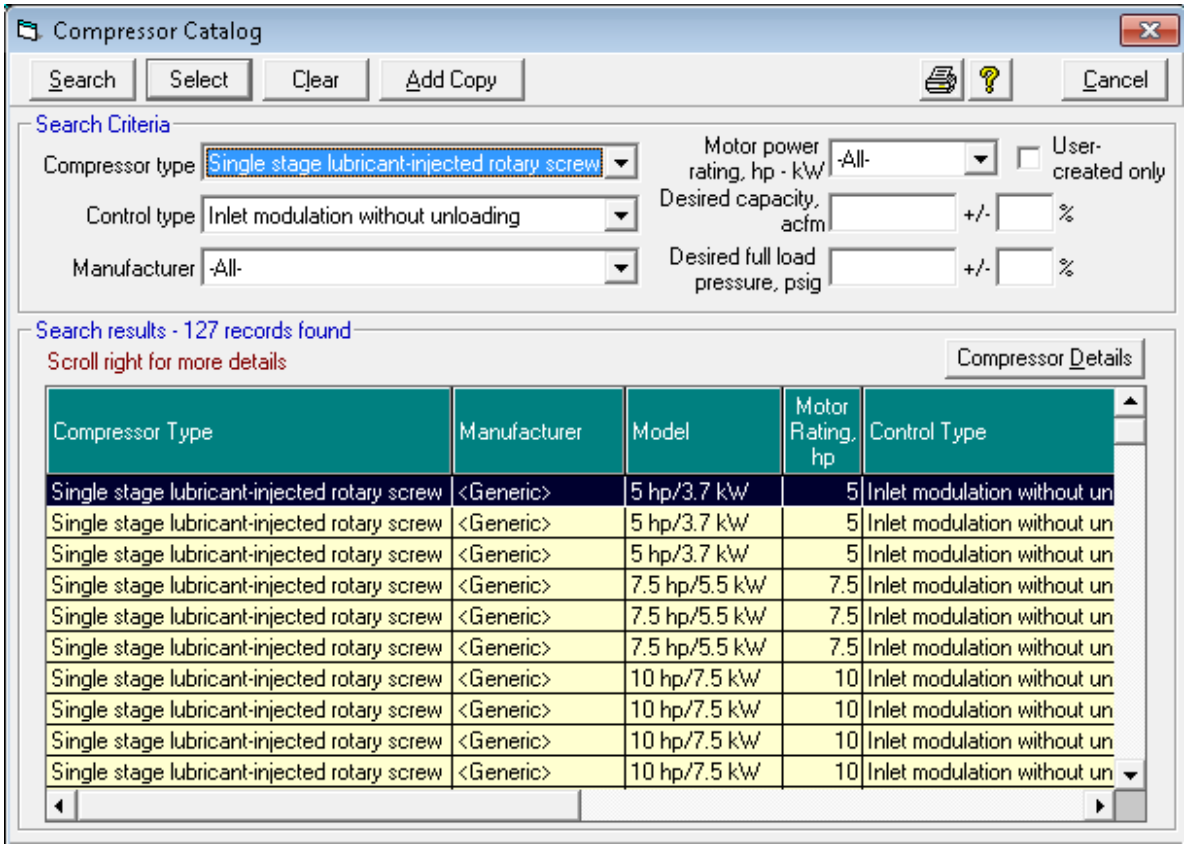


Figure 68: AIRMaster+ Compressor Module Compressor Catalog (15)

For example, if one wanted to add a 150 hp, single stage lubricant injected rotary screw compressor using modulating controls, the search and results would be as shown in Figure 69. For this project, an entry was made for each of the facility's compressors. Each compressor was chosen from a generic entry, which closely resembled the actual compressor specifications. The chosen compressors are shown in Figure 70 below.

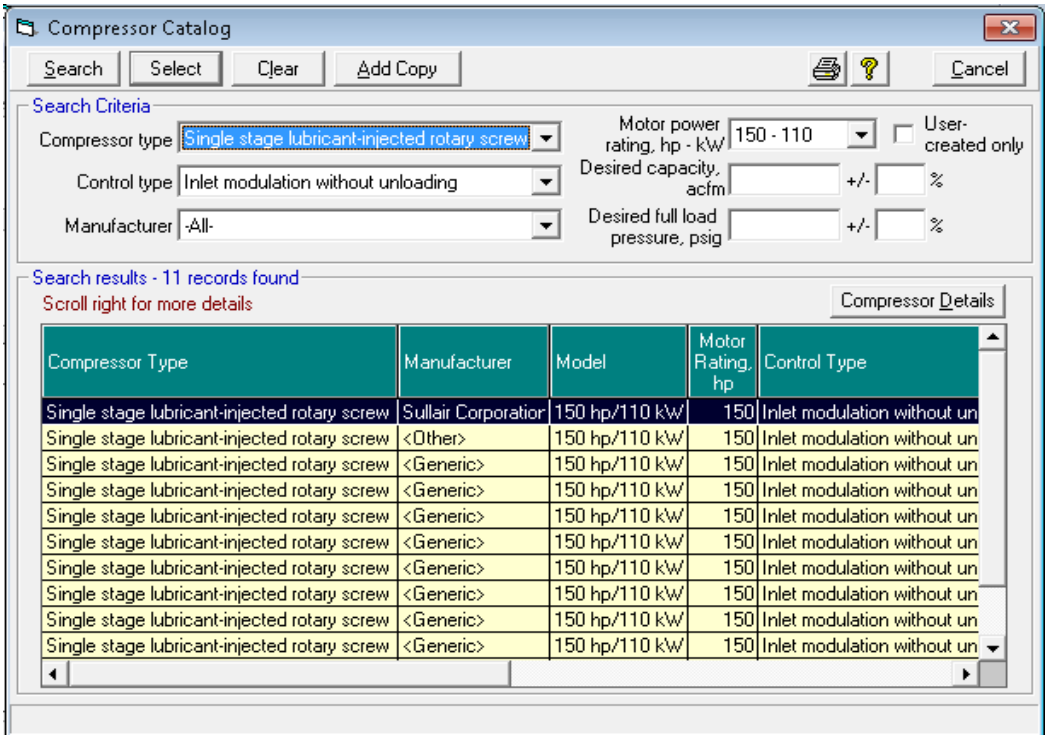


Figure 69: AIRMaster+ Compressor Catalog Search (15)

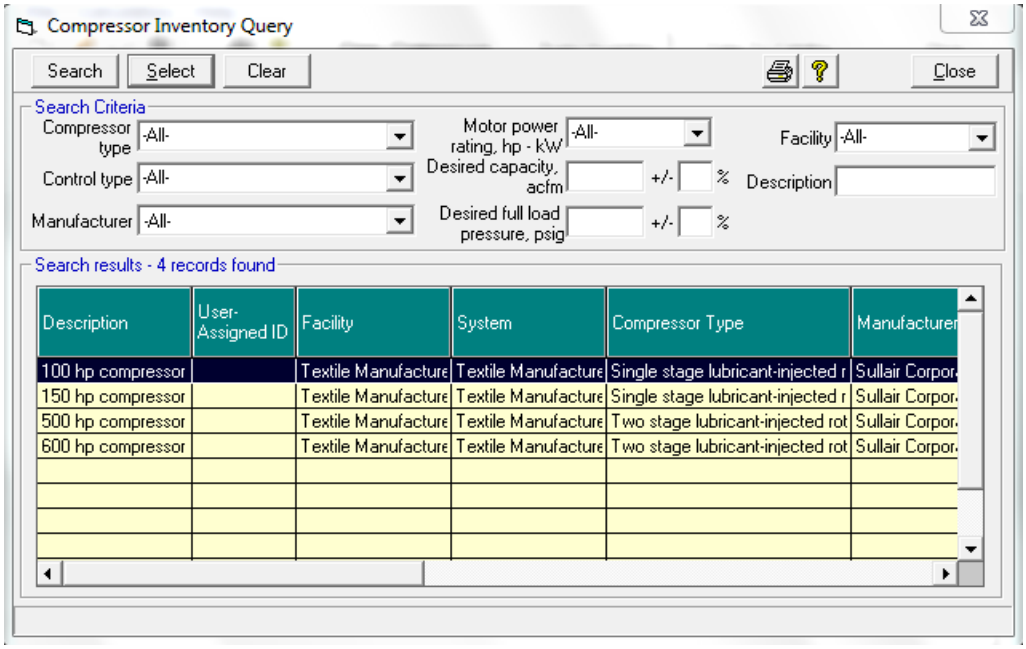


Figure 70: Compressor Inventory (15)

Once the compressors are chosen, the individual compressor controls and performance can be modified further to match the actual compressor. The performance curves for each compressor are shown below. The performance profile indicates the percent full load power the compressor uses at a certain percent capacity.

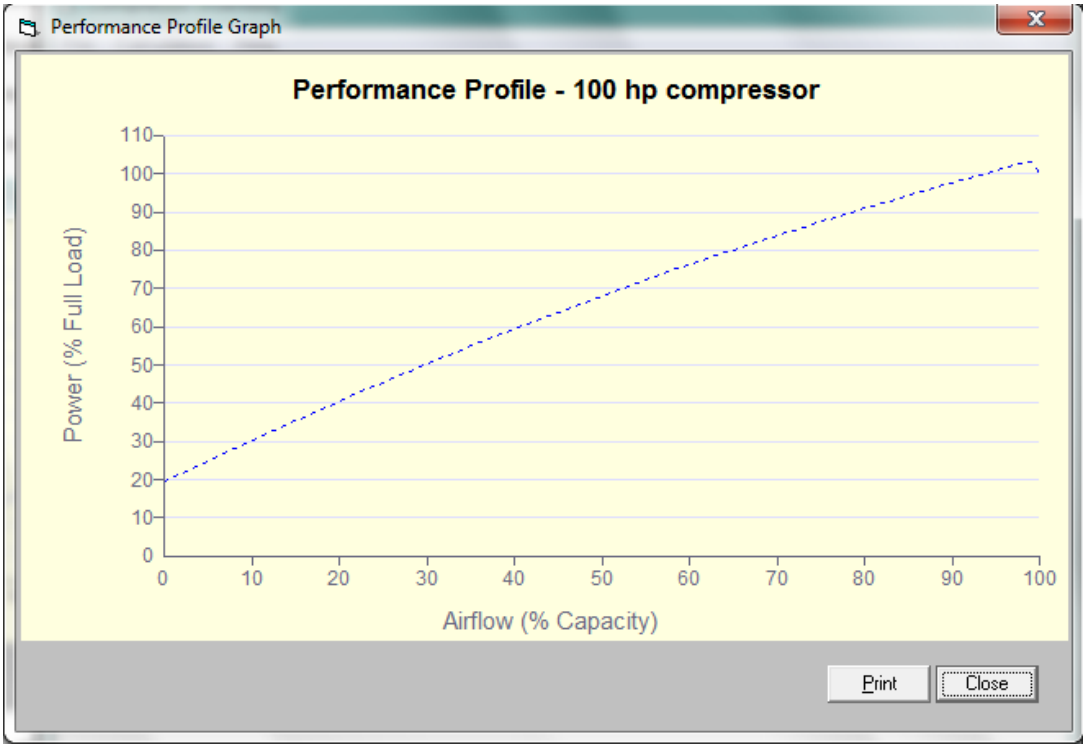


Figure 71: 100 hp Compressor Performance Profile (15)

Figure 71 is indicative of an air compressor using load/unload controls. Because the compressor is allowed to unload, one can expect to see a low percentage of full load power when no compressed air is being produced.

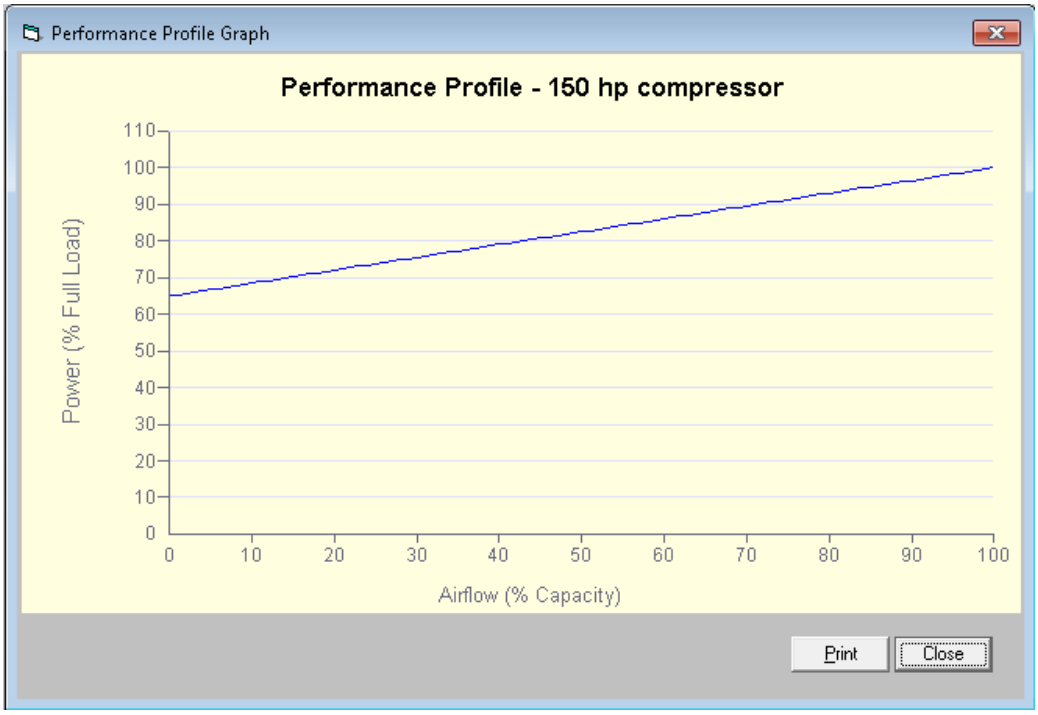


Figure 72: 150 hp Compressor Performance Profile (15)

The performance curve in Figure 72 is a classic curve seen with compressors utilizing modulation controls. The compressor tends to use a high percentage of its full load power even when it is producing no compressed air. Because the 150 hp, 500 hp, and 600 hp compressors all use modulating controls, it is reasonable that their performance curves would be similar.

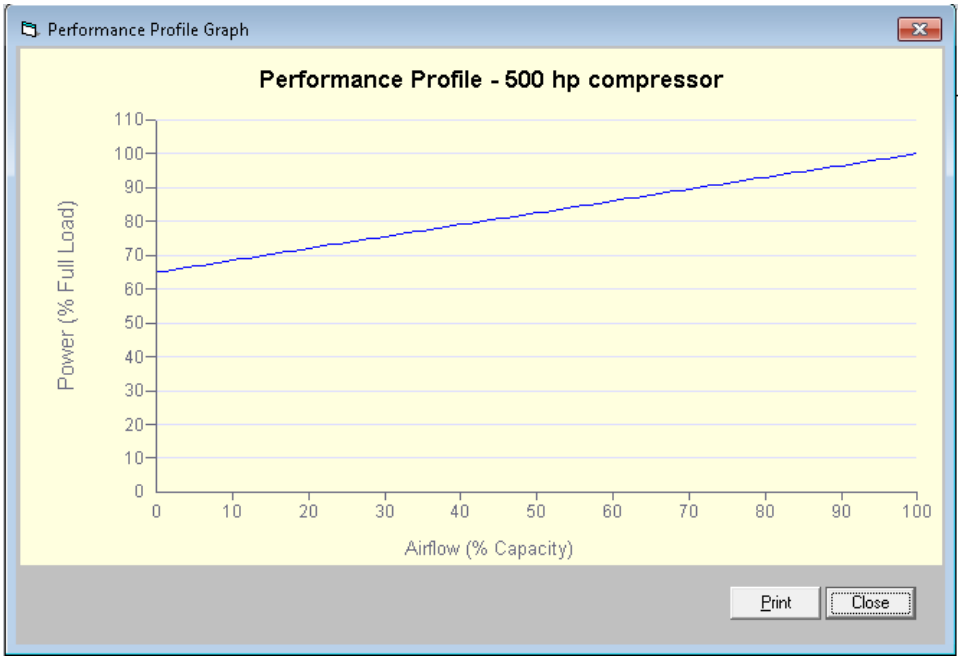


Figure 73: 500 hp Compressor Performance Profile (15)

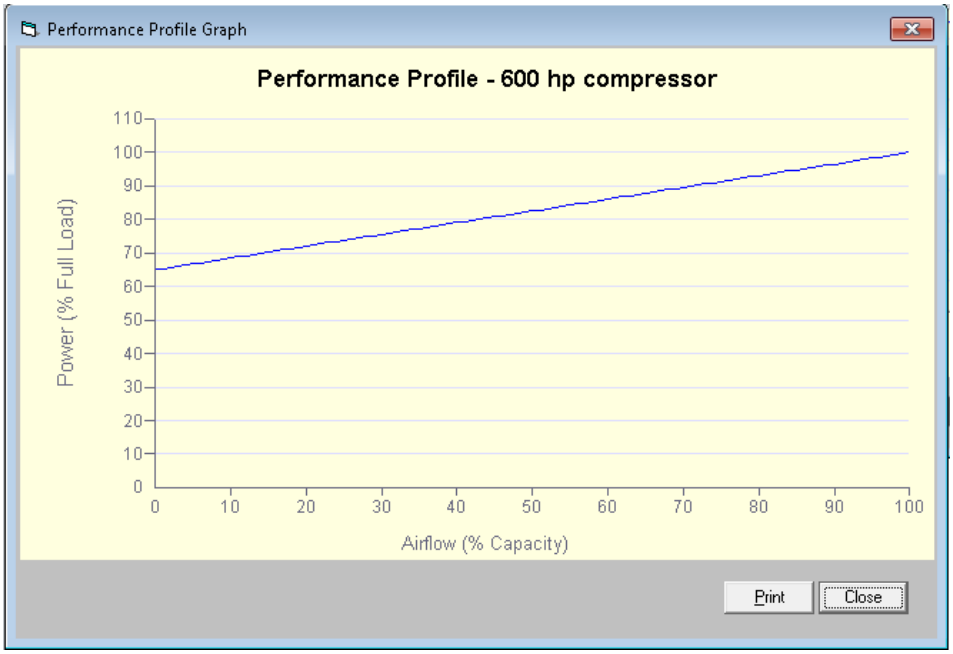


Figure 74: 600 hp Compressor Performance Profile (15)

5.1.5.1 Compressor Pressure Performance Control Points

Another important aspect to setting up an adequate model is to specify the compressor pressure control range. Each compressor has a pressure band, which it adheres to in order to meet demand. This compressor system likely cascades the compressors with different pressure performance points for each compressor.

The 600 hp compressor and the 100 hp compressor currently operate together to meet compressed air demand. The 600 hp compressor is the base load compressor, and the 100 hp is the swing compressor to meet any additional demand. Using modulating controls, the 600 hp compressor controls between 110 psig and 120 psig, where the compressor is fully loaded when the system air pressure is 110 psig or below, and is completely modulated when the system air pressure reaches 120 psig. Thus, if the system air pressure is below 110 psig, the 600 hp compressor is fully loaded all the time. The 100 hp compressor likely has a control band from approximately 100 psig to 113 psig. At 113 psig, the compressor unloads, which is called the cut-out pressure. When the pressure drops under 100 psig, the compressor cuts back in at maximum power. To further illustrate the operation of the compressors; if the system air pressure is 108 psig, the 600 hp compressor will be fully loaded, and the 100 hp compressor will be partly loaded. Should the system air pressure drop below 100 psig, both compressors would be fully loaded to meet increased demand.

The performance points entered into AIRMaster+ for each compressor are as follows. The 600 and 500 hp compressors are fully loaded at 110 psig, and fully modulated at 120 psig. As the trim compressor, the 100 hp compressor has a cut-in pressure of 100 psig, and a cut-out pressure of 113 psig. The 150 hp compressor is the base compressor during operation on Saturdays and Sundays, therefore the compressor is fully loaded at 110 psig, and fully modulated at 120 psig.

5.1.6 LogTool

The next step is to add operation profiles for each compressor. This is to indicate when and how the compressors operate. Using data acquired by data logging, the user can indicate how each compressor operates within the compressed air system.

The first step is to input logged data into a program called LogTool, which is a public domain tool made available from SBW Consulting, Inc. and the Compressed Air Challenge. The program allows the user to create a daytype, which is collected data averaged over one hour periods for an entire day. The LogTool opening screen is shown below.

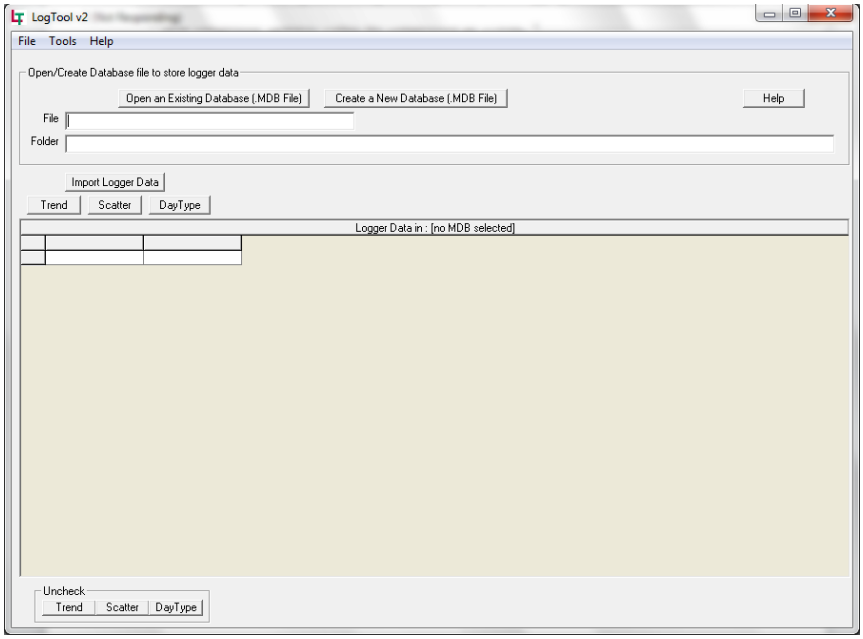


Figure 75: Log Tool Opening Screen (16)

From the opening screen the user should create a new database. The database allows data loggers to be uploaded, and daytypes to be created. The amperage data from the facility's four compressors were uploaded to the database. Three daytypes, including Production, Saturdays, and Sundays, were created for each compressor.

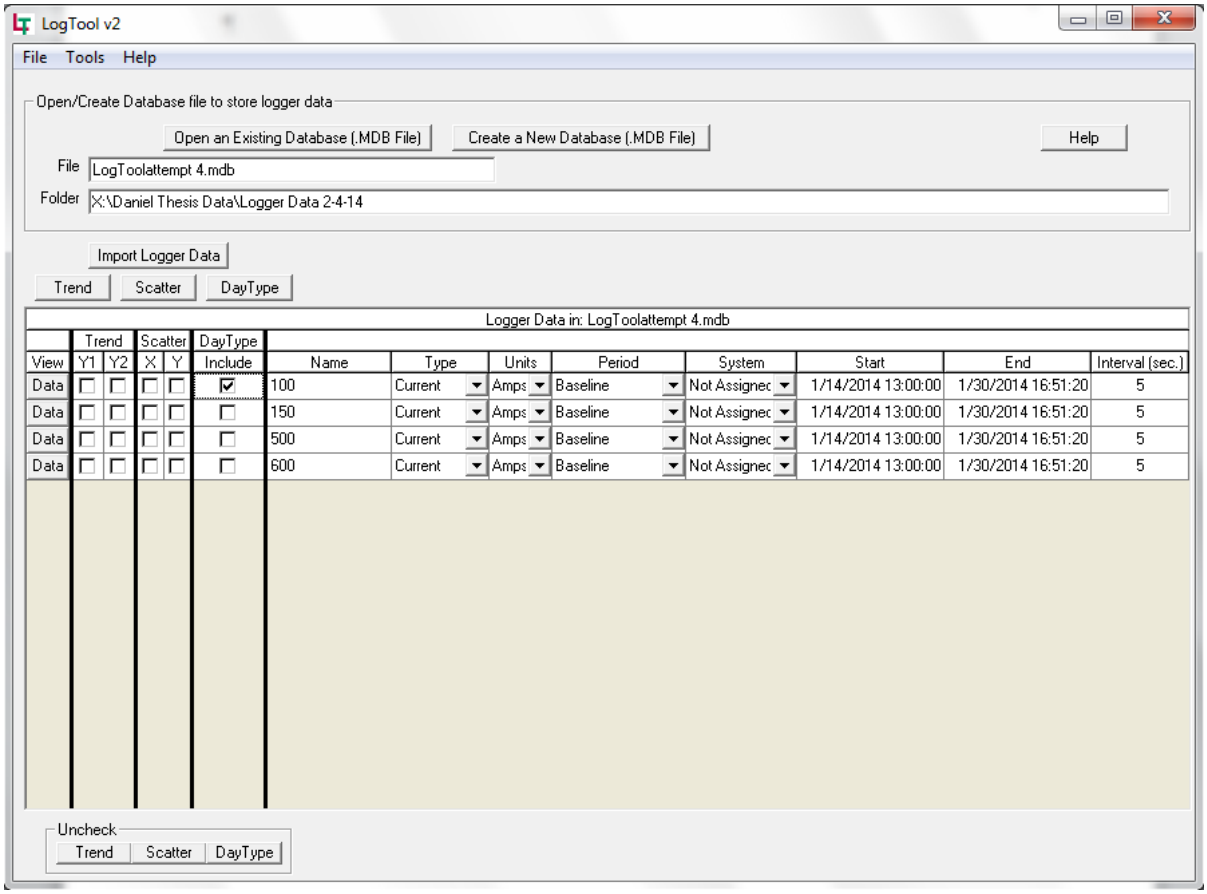
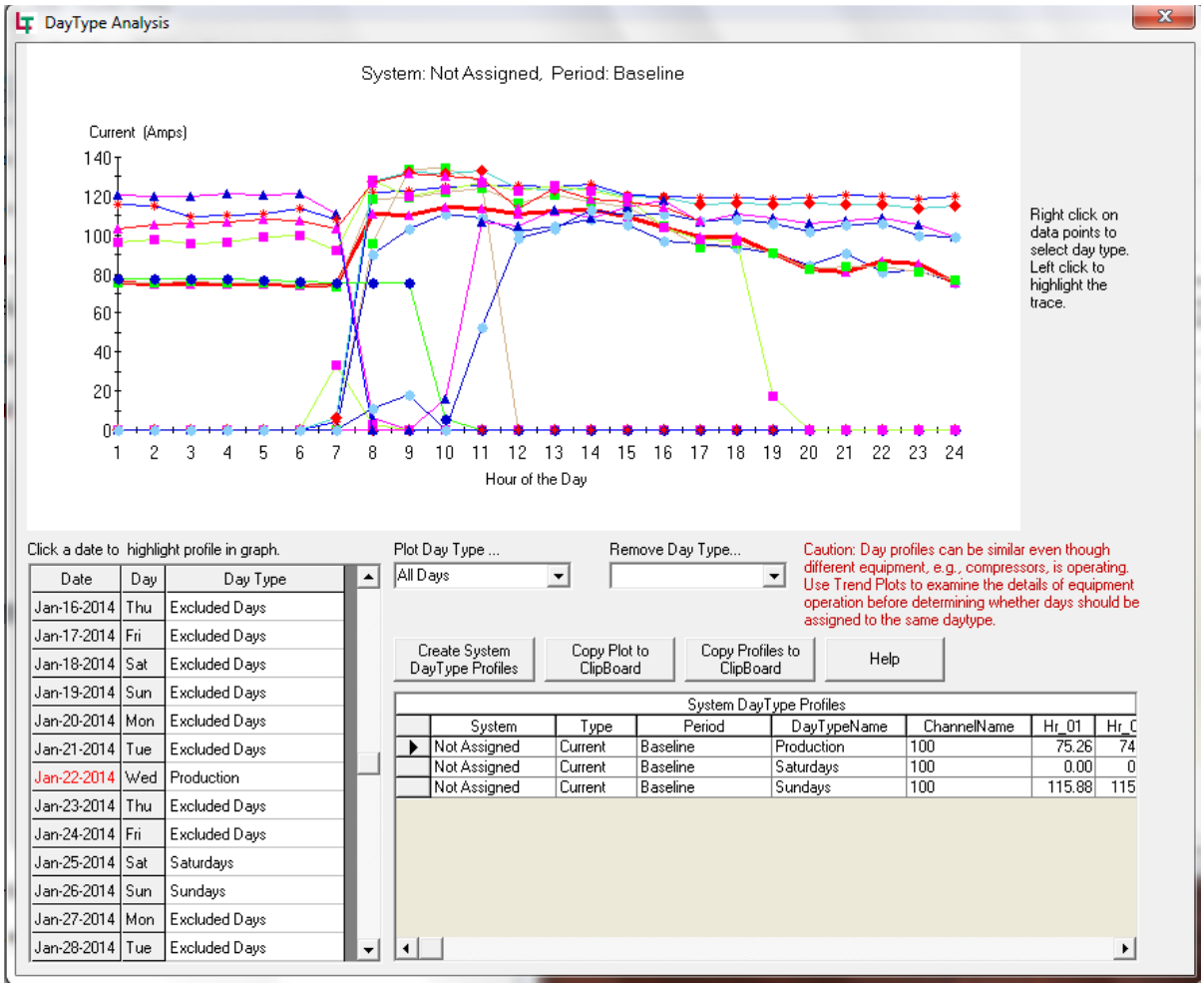

LogTool v2
File Tools Help
Open/Create Database file to store logger data
Open an Existing Database (.MDB File)
Create a New Database (.MDB File)
Help
File LogToolattempt 4.mdb
Folder X:\Daniel Thesis Data\Logger Data 2-4-14
Import Logger Data
Trend Scatter DayType
Logger Data in: LogToolattempt 4.mdb
Trend Scatter DayType
View Y1 Y2 X Y Include Name Type Units Period System Start End Interval (sec.)
Data 100 Current Amps Baseline Not Assignec 1/14/2014 13:00:00 1/30/2014 16:51:20 5
Data 150 Current Amps Baseline Not Assignec 1/14/2014 13:00:00 1/30/2014 16:51:20 5
Data 500 Current Amps Baseline Not Assignec 1/14/2014 13:00:00 1/30/2014 16:51:20 5
Data 600 Current Amps Baseline Not Assignec 1/14/2014 13:00:00 1/30/2014 16:51:20 5
Uncheck
Trend Scatter DayType

Figure 76: LogTool Imported Loggers (16)

To create a daytype for a compressor, click check box next to the compressor for a particular compressor and then press the DayType button above. The program will output the average amperage hourly for each day of data collection. The user must choose a day that most resembles the daytype that is desired. For example, the user would choose a typical operating day for a production daytype, and an average Saturday and Sunday for those daytypes respectively. Daytypes for the 100 hp compressor will be chosen first, followed by the 150 hp compressor, 500 hp compressor, and 600 hp compressor. Wednesday January 22nd was

chosen as a Production day. The chosen dates for the daytypes for Saturdays and Sundays are January 25th and January 26th respectively.


DayType Analysis
System: Not Assigned, Period: Baseline
Current (Amps)
Hour of the Day
Right click on data points to select day type. Left click to highlight the trace.
Click a date to highlight profile in graph.
Plot Day Type ...
All Days
Remove Day Type...
Caution: Day profiles can be similar even though different equipment, e.g., compressors, is operating. Use Trend Plots to examine the details of equipment operation before determining whether days should be assigned to the same daytype.
Create System DayType Profiles
Copy Plot to ClipBoard
Copy Profiles to ClipBoard
Help
System DayType Profiles

Date	Day	Day Type
Jan-16-2014	Thu	Excluded Days
Jan-17-2014	Fri	Excluded Days
Jan-18-2014	Sat	Excluded Days
Jan-19-2014	Sun	Excluded Days
Jan-20-2014	Mon	Excluded Days
Jan-21-2014	Tue	Excluded Days
Jan-22-2014	Wed	Production
Jan-23-2014	Thu	Excluded Days
Jan-24-2014	Fri	Excluded Days
Jan-25-2014	Sat	Saturdays
Jan-26-2014	Sun	Sundays
Jan-27-2014	Mon	Excluded Days
Jan-28-2014	Tue	Excluded Days

System	Type	Period	DayTypeName	ChannelName	Hr_01	Hr_0
Not Assigned	Current	Baseline	Production	100	75.26	74
Not Assigned	Current	Baseline	Saturdays	100	0.00	0
Not Assigned	Current	Baseline	Sundays	100	115.88	115

Figure 77: 100 hp Compressor Daytypes (16)

The user must then copy profiles to the clipboard, and then paste into a spreadsheet. All daytpes will be pasted into the same spreadsheet to easily transfer the daytpes into AIRMaster+. Figure 78 is a plot of the three daytypes for the 100 hp compressor.

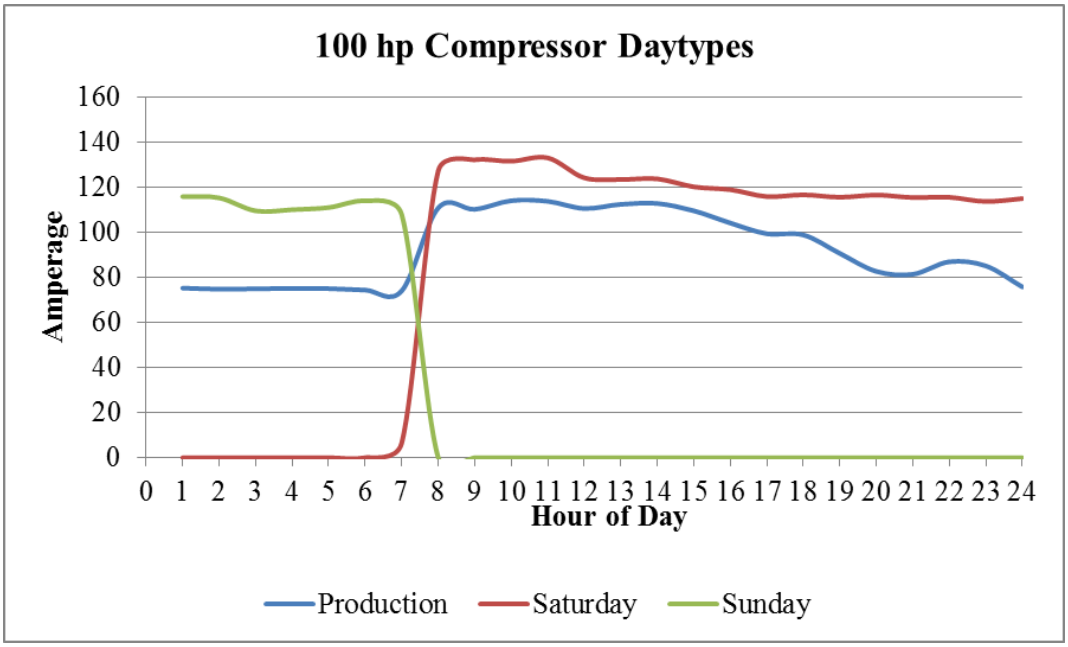


Figure 78: Plot of 100 hp Compressor Daytypes

Since the 150 hp compressor rarely runs on an average production day, the production daytype was chosen from a day in which the compressor did not run. The compressor does run on Saturdays and Sundays, thus appropriate days were chosen for the daytype. Friday, January 24th was chosen for the production daytype, followed by January 25th and 26th for the Saturday and Sunday daytypes. In Figure 79, the chosen daytypes for the 150 hp compressor can be viewed. Figure 80 is a plot of the 150 hp daytypes.

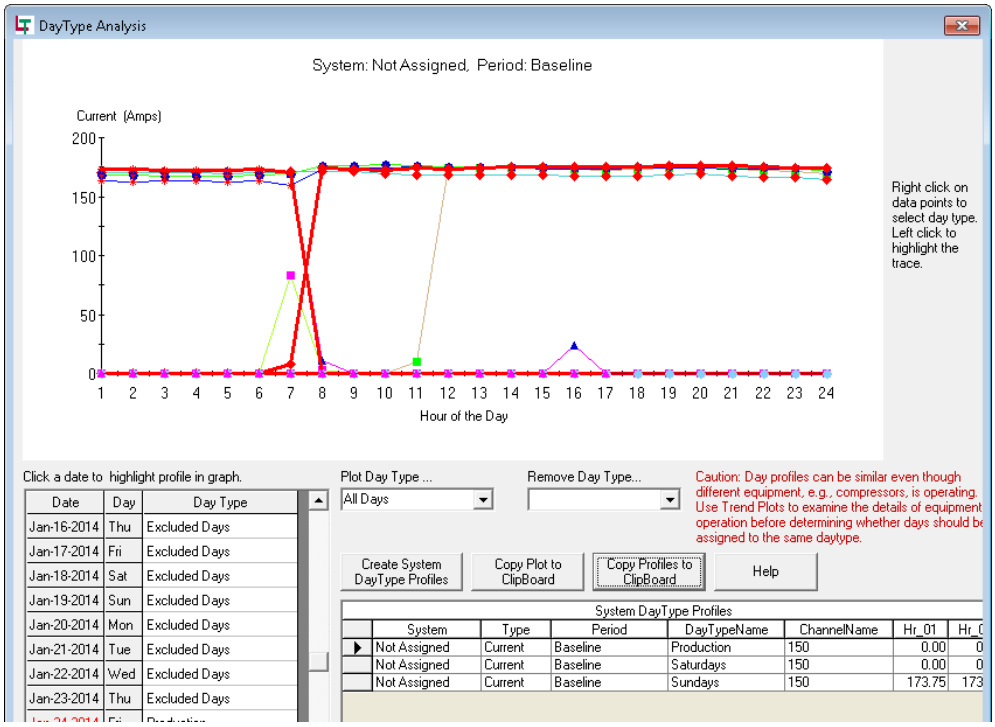

DayType Analysis
System: Not Assigned, Period: Baseline
Current (Amps)
200
150
100
50
0
1 2 3 4 5 6 7 8 9 10 11 12 13 14 15 16 17 18 19 20 21 22 23 24
Hour of the Day
Right click on data points to select day type. Left click to highlight the trace.
Click a date to highlight profile in graph.
Date
Day
Day Type
Jan-16-2014 Thu Excluded Days
Jan-17-2014 Fri Excluded Days
Jan-18-2014 Sat Excluded Days
Jan-19-2014 Sun Excluded Days
Jan-20-2014 Mon Excluded Days
Jan-21-2014 Tue Excluded Days
Jan-22-2014 Wed Excluded Days
Jan-23-2014 Thu Excluded Days
Plot Day Type ...
All Days
Remove Day Type...
Caution: Day profiles can be similar even though different equipment, e.g., compressors, is operating. Use Trend Plots to examine the details of equipment operation before determining whether days should be assigned to the same daytype.
Create System DayType Profiles
Copy Plot to ClipBoard
Copy Profiles to ClipBoard
Help
System DayType Profiles
System Type Period DayTypeName ChannelName Hr_01
Not Assigned Current Baseline Production 150 0.00
Not Assigned Current Baseline Saturdays 150 0.00
Not Assigned Current Baseline Sundays 150 173.75

Figure 79: LogTool 150 hp Compressor Daytypes (16)

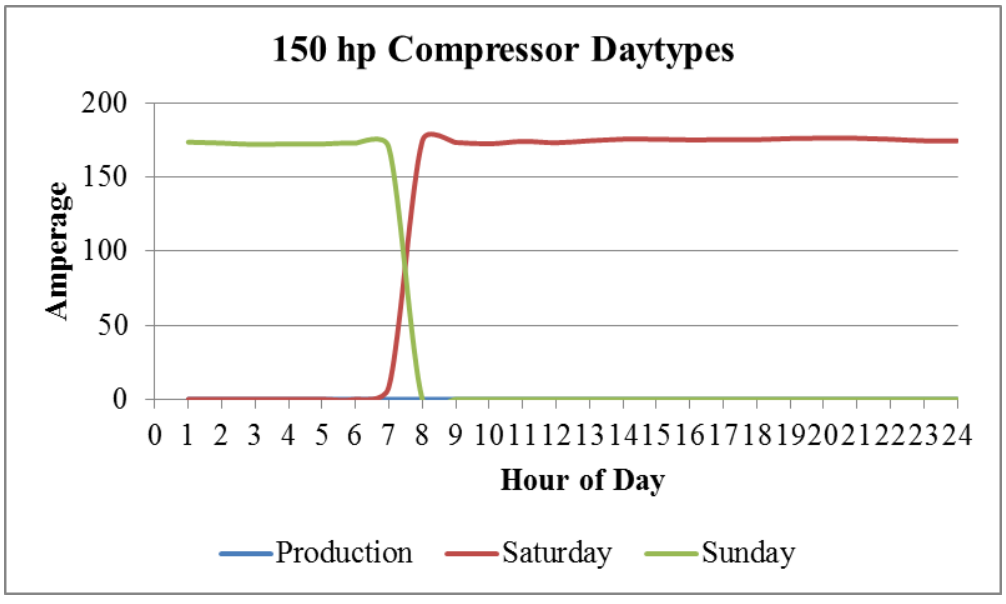


Figure 80: Plot of 150 hp Compressor Daytypes

The 500 hp compressor is not a primary compressor, and does not ordinarily operate. Generally the 600 hp compressor is the base compressor, operating in conjunction with a smaller compressor. Thus, the 500 hp compressor is considered off at all times during modeling simulations. Nevertheless, daytypes were created, but the average amperage values for each daytype were zero. As seen in Figure 81, the 500hp compressor rarely turned on, but operated at close to full load on 3 days. The effect of setting the daytypes to zero for the 500 hp compressor is that AIRMaster+ will model the 600 hp compressor as the base compressor.

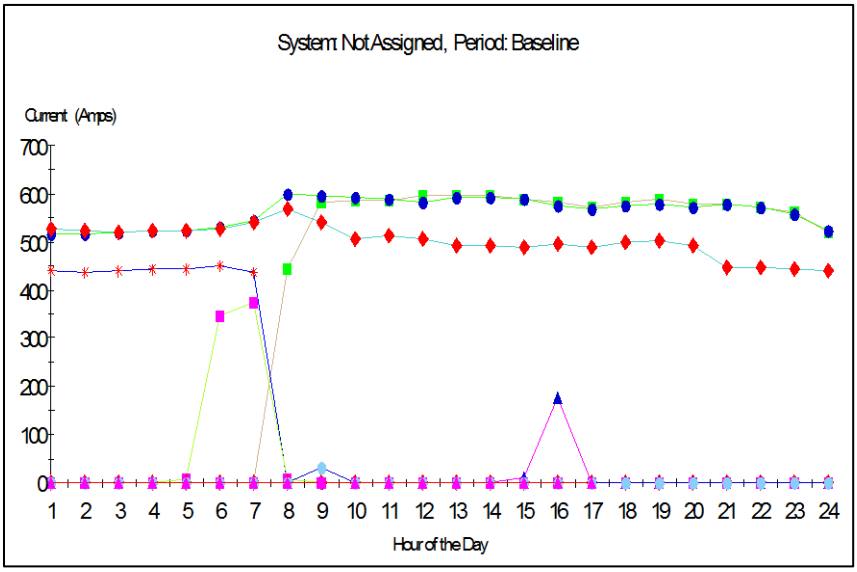


Figure 81: 500 hp LogTool Daytypes Plot (16)

The 600 hp compressor operates all hours from Monday to Friday, and for a few hours into the morning on Saturdays. Thursday January 26th was chosen as a typical production day. January 25th and January 26th were chosen to represent typical Saturdays and Sundays.

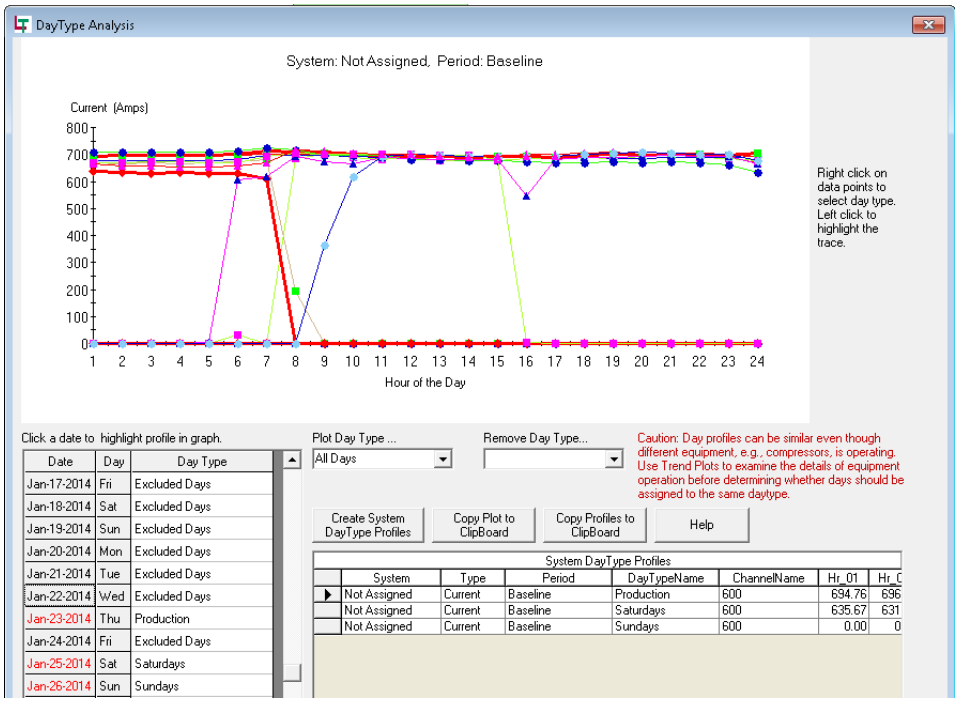

DayType Analysis
System: Not Assigned, Period: Baseline
Current (Amps)
800
700
600
500
400
300
200
100
0
1 2 3 4 5 6 7 8 9 10 11 12 13 14 15 16 17 18 19 20 21 22 23 24
Hour of the Day
Right click on data points to select day type. Left click to highlight the trace.
Click a date to highlight profile in graph.
Date | Day | Day Type
Jan-17-2014 | Fri | Excluded Days
Jan-18-2014 | Sat | Excluded Days
Jan-19-2014 | Sun | Excluded Days
Jan-20-2014 | Mon | Excluded Days
Jan-21-2014 | Tue | Excluded Days
Jan-22-2014 | Wed | Excluded Days
Jan-23-2014 | Thu | Production
Jan-24-2014 | Fri | Excluded Days
Jan-25-2014 | Sat | Saturdays
Jan-26-2014 | Sun | Sundays
Plot Day Type ...
All Days
Remove Day Type...
Caution: Day profiles can be similar even though different equipment, e.g., compressors, is operating. Use Trend Plots to examine the details of equipment operation before determining whether days should be assigned to the same daytype.
Create System DayType Profiles
Copy Plot to ClipBoard
Copy Profiles to ClipBoard
Help
System DayType Profiles
System | Type | Period | DayTypeName | ChannelName | Hr_01 | Hr_0
Not Assigned | Current | Baseline | Production | 600 | 694.76 | 696
Not Assigned | Current | Baseline | Saturdays | 600 | 635.67 | 631
Not Assigned | Current | Baseline | Sundays | 600 | 0.00 | 0

Figure 82: 600 hp Compressor LogTool Daytypes (16)

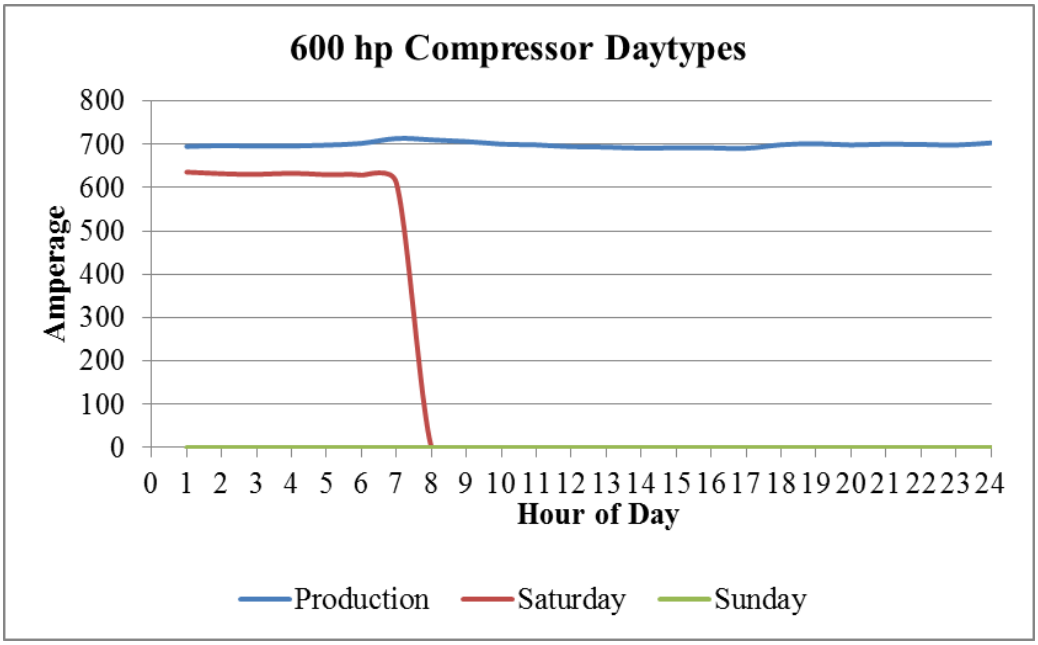


Figure 83: 600 hp Compressor Daytypes

5.1.7 Profile Module

After the daytypes have been have been defined, they can be applied to AIRMaster+. The voltage and average amperage for each compressor for all three daytypes are pasted into AIRMaster+ to create the system profile.

In the System Profiles module, first the daytype is chosen, the cascade order is or an on/off position is chosen, and then the voltage and averaged amperage data is entered. Figure 84 is the system profile for the production daytype. .

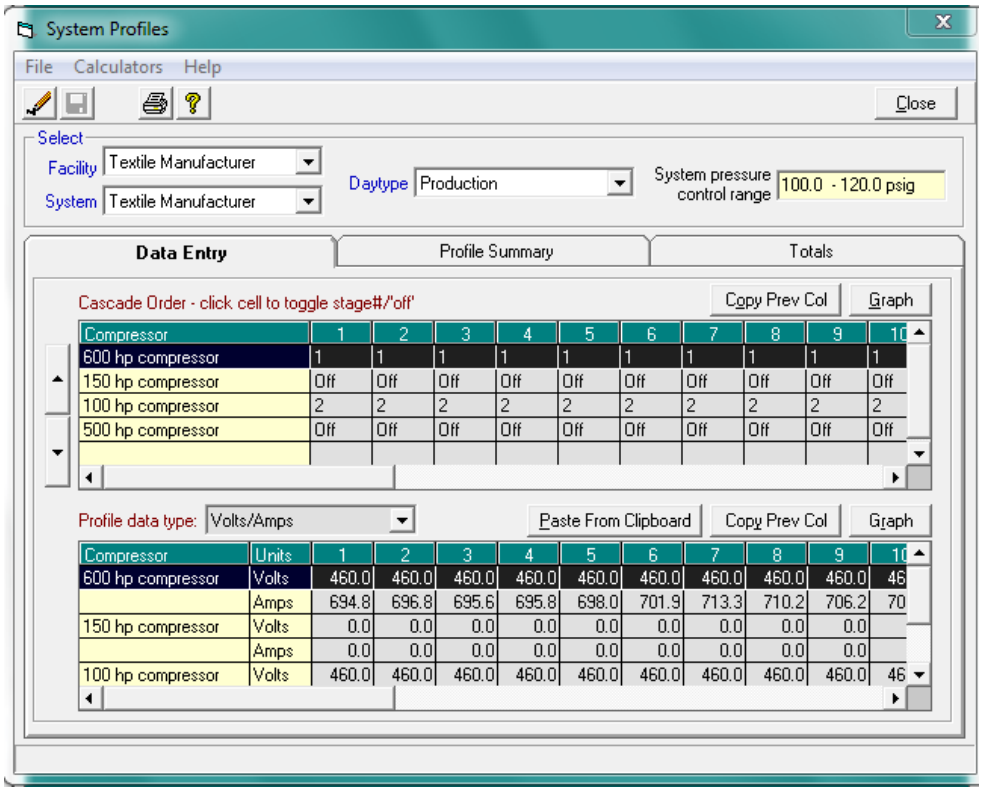

System Profiles
File Calculators Help
Close
Select
Facility Textile Manufacturer
System Textile Manufacturer
Daytype Production
System pressure control range 100.0 - 120.0 psig
Data Entry
Profile Summary
Totals
Cascade Order - click cell to toggle stage#/'off'
Copy Prev Col
Graph
Compressor 1 2 3 4 5 6 7 8 9 10
600 hp compressor 1 1 1 1 1 1 1 1 1 1
150 hp compressor Off Off Off Off Off Off Off Off Off Off
100 hp compressor 2 2 2 2 2 2 2 2 2 2
500 hp compressor Off Off Off Off Off Off Off Off Off Off
Profile data type: Volts/Amps
Paste From Clipboard
Copy Prev Col
Graph
Compressor Units 1 2 3 4 5 6 7 8 9 10
600 hp compressor Volts 460.0 460.0 460.0 460.0 460.0 460.0 460.0 460.0 460.0 46
Amps 694.8 696.8 695.6 695.8 698.0 701.9 713.3 710.2 706.2 70
150 hp compressor Volts 0.0 0.0 0.0 0.0 0.0 0.0 0.0 0.0 0.0
Amps 0.0 0.0 0.0 0.0 0.0 0.0 0.0 0.0 0.0
100 hp compressor Volts 460.0 460.0 460.0 460.0 460.0 460.0 460.0 460.0 460.0 46

Figure 84: System Profile Module Production Daytype (15)

The system profile data for the production daytype is plotted below. The plots include volumetric air flow (ACFM), and power (kW). Figure 85 shows the volumetric flow by the hour for the average production day. The peak air flow is approximately 3103 ACFM at 8 AM.

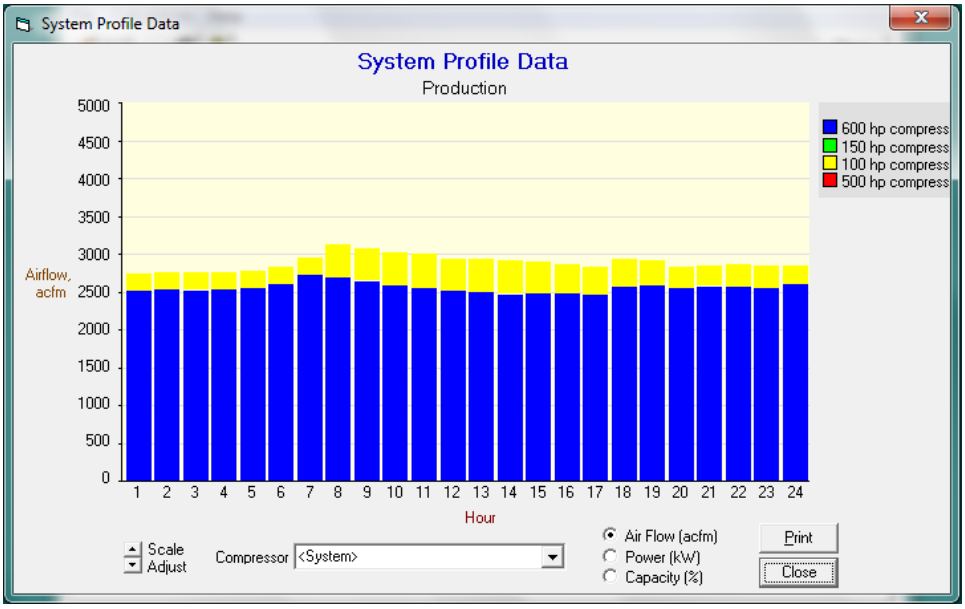


Figure 85: Production Profile Compressor Volumetric Flow (15)

Figure 86 is a plot of compressor power by the hour of day for the average production day. The peak power occurs at 8 am, with a power of 580.5 kW.

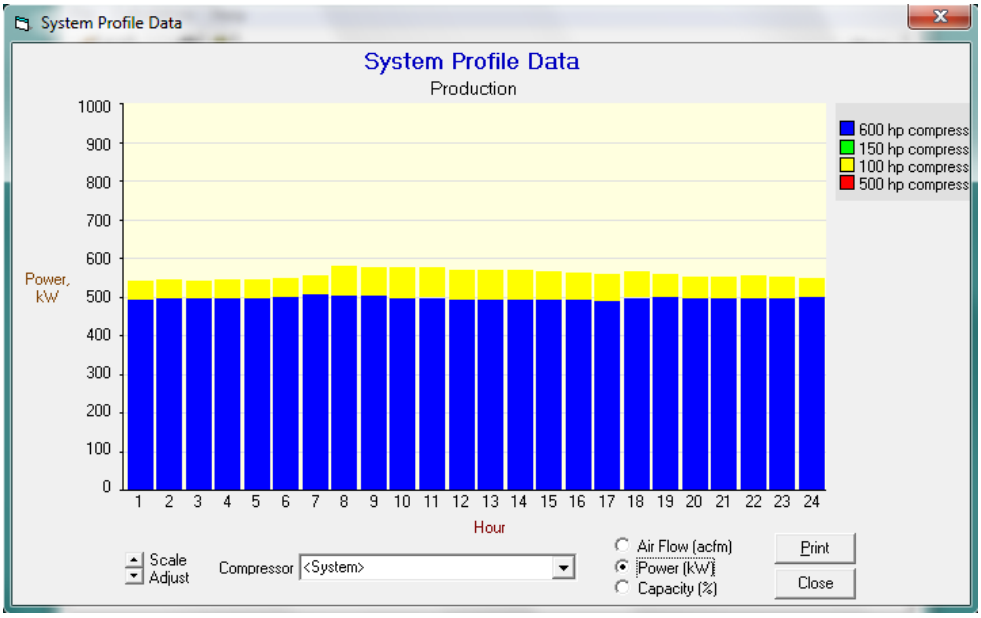


Figure 86: Figure 85: Production Profile Compressor Power (15)

The next daytype to consider is the Saturday daytype. Two compressors generally operate on Saturdays, the 100 hp compressor and the 150 hp compressor. Thus, less power is expected to be required for this daytype, and the compressed air demand will be much lower. As seen in Figure 87, the 600 hp compressor operates briefly in the early hours of the daytype, but at around 8 am the 100 hp compressor and the 150 hp compressor begin to supply compressed air to satisfy demand. Although the 600 hp compressor is operating from hour 1 to hour 7, the compressor capacity is low. This indicates that the compressor is operating inefficiently, for a modulating compressor still requires a high percentage of its power at lower capacity.

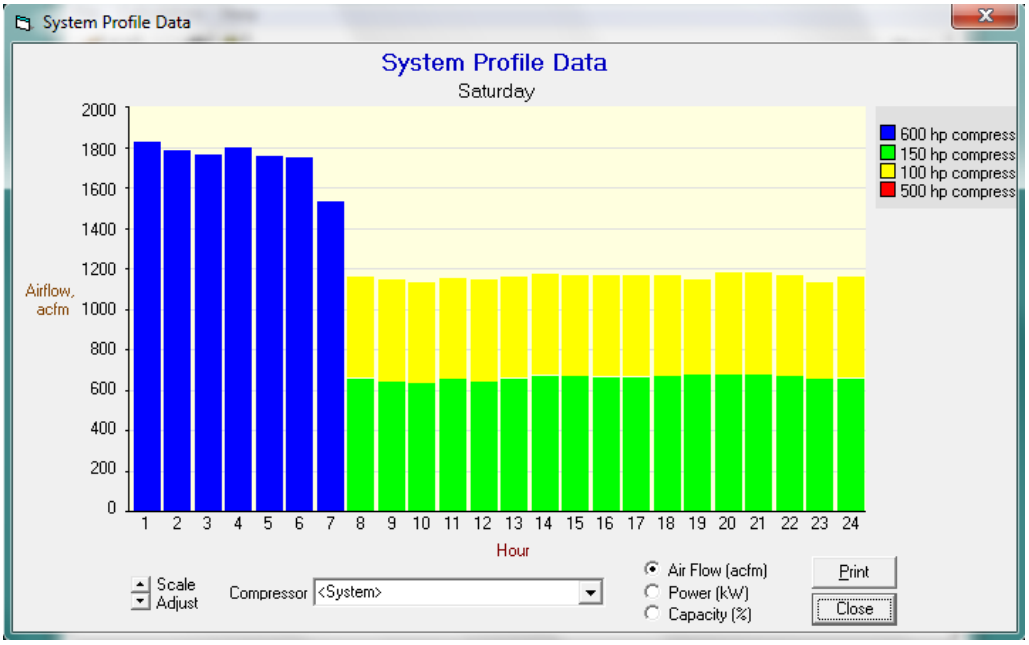


Figure 87: Saturday Profile Volumetric Flow (15)

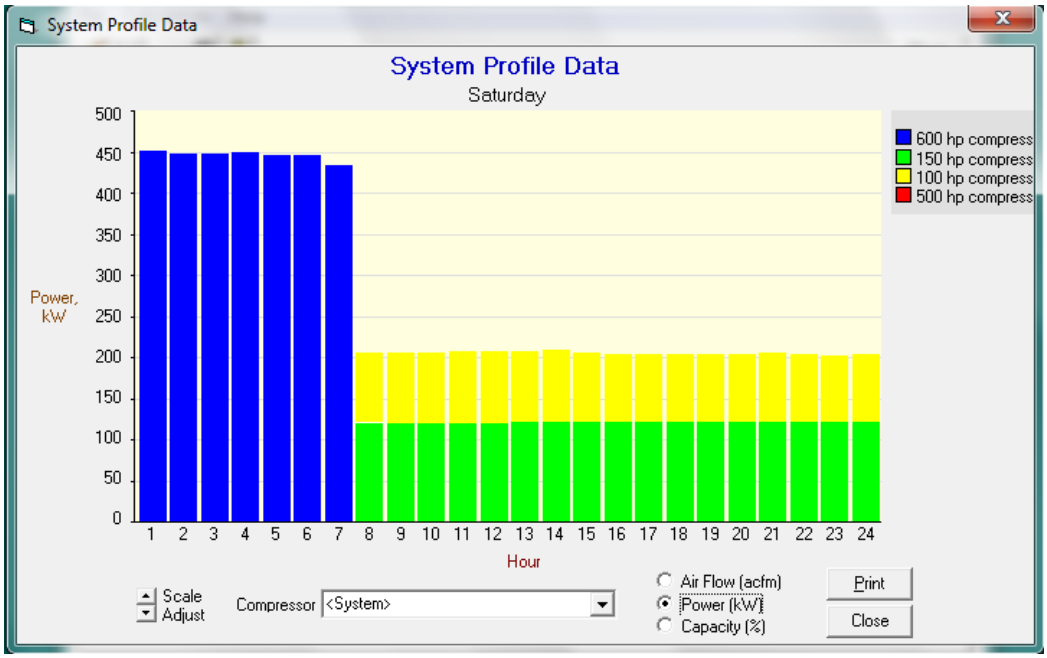


Figure 88: Saturday Profile Data Power (15)

Figure 88 is a plot of power over the Saturday daytype. As expected, the power is highest when the 600 hp compressor is operating.

Sundays are the final daytype to be profiled. After inputting data, the profile indicates that the 100 hp compressor and the 150 hp compressor are the only compressors to operate, and only from hour 1 to hour 7. Figure 89 is a plot of air flow for the Sunday system profile. The peak air flow is just under 1200 ACFM. After hour 7, the compressors turn off, and the system compressed air demand remains at 0 ACFM until production resumes on Mondays.

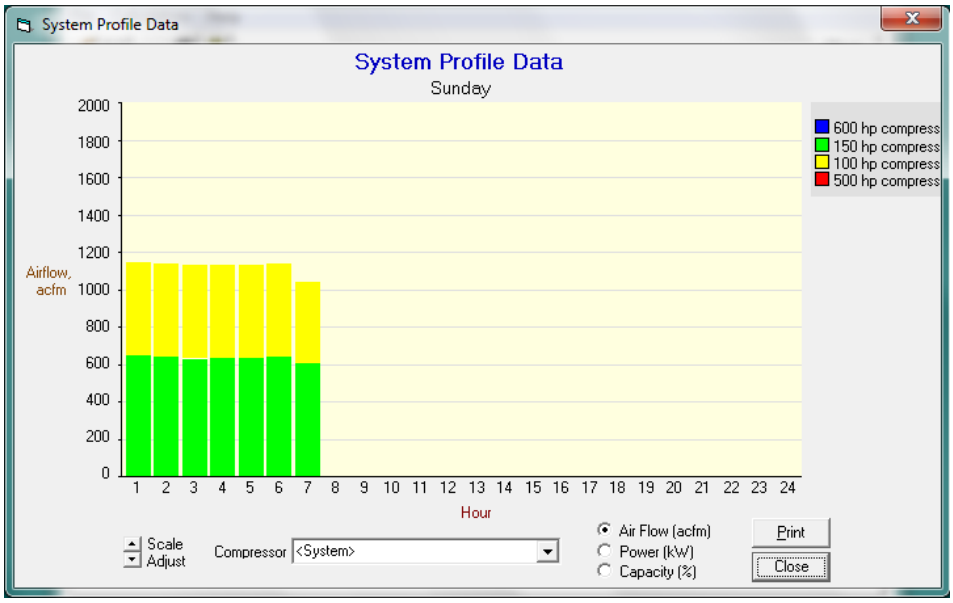


Figure 89: Sunday Profile Volumetric Flow (15)

Figure 90 illustrates the compressor power for each hour of the Sunday profile. The peak power is just over 200 kW.

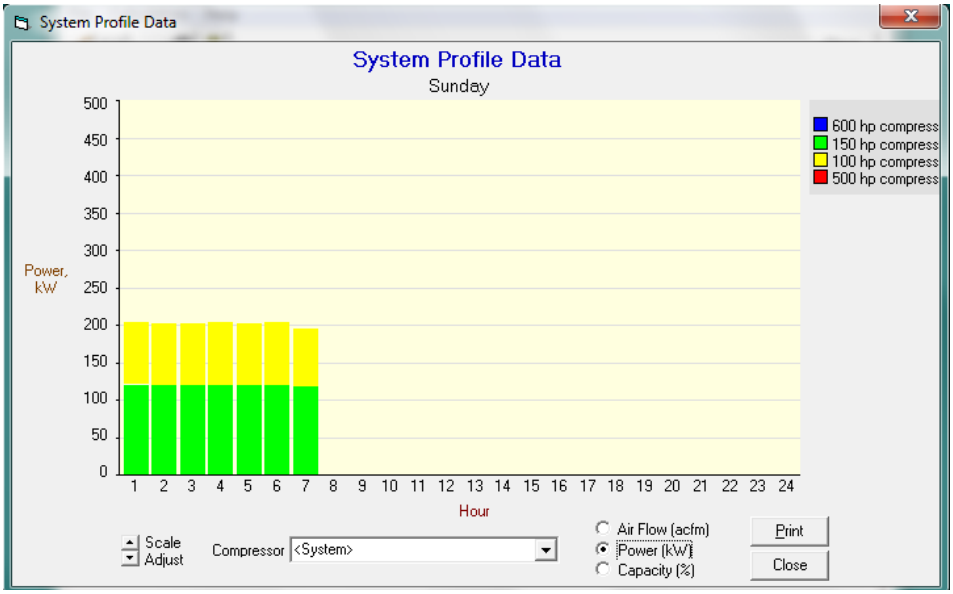


Figure 90: Sunday Profile Power (15)

The profile totals, generated in AIRMaster+, for average airflow (ACFM), peak demand (kW), annual energy (kWh), and annual energy cost are located in Table 6: System Profile Totals below.

Table 6: System Profile Totals

Daytype	Total OpHrs	Avg Airflow, acfm	Avg Airflow, %Cs.	Peak Demand, kW	Load Factor, %	Annual Energy, kWh	Annual Energy Cost, $
Production	6,000	2,857	42.6	580.5	48.0	3,349,180	143,680
Saturday	1,200	1,327	19.8	450.8	23.6	330,162	14,164
Sunday	350	1,119	16.7	203.2	17.3	70,507	3,025
Total	7,550	2,533	37.8	580.5	42.7	3,749,981	160,869

5.2 Energy Efficiency Measures

AIRMaster+ calculates the savings from several energy efficiency measures. On the demand side of the compressed air system, the user can calculate savings from reducing the system air leak load, improving end use efficiency, and reduce system air pressure. The software also offers a number of supply side efficiency measures. Savings can be calculated from using unloading controls, adjusting cascading set points, the use of automatic sequencers, reducing compressor run time, and adding primary receiver volume.

This project will use AIRMaster+ to simulate savings from the controlling pressure and compressor operation with the use of an automatic sequencer, and the reduction of air leaks. The order of implementation of these recommendations does matter. Decreasing the system air pressure will naturally decrease the leak load.

To validate the AIRMaster+ simulation, a simplified model will first be made to calculate the savings from reducing system air pressure, and then savings will be calculated by an alternate method.

5.2.1 Reduce System Air Pressure AIRMaster+ Validation

A simplified AIRMaster+ model, using only the facility's 600 hp compressor, must first be created. The utility information is the same as described in section 5.1.2. Using LogTool, the compressor profile for the production daytype was created then pasted into AIRMaster+. The compressor pressure control range is modeled to be 103 psig at full load, and 120 psig at no load. The compressor profile is shown below.

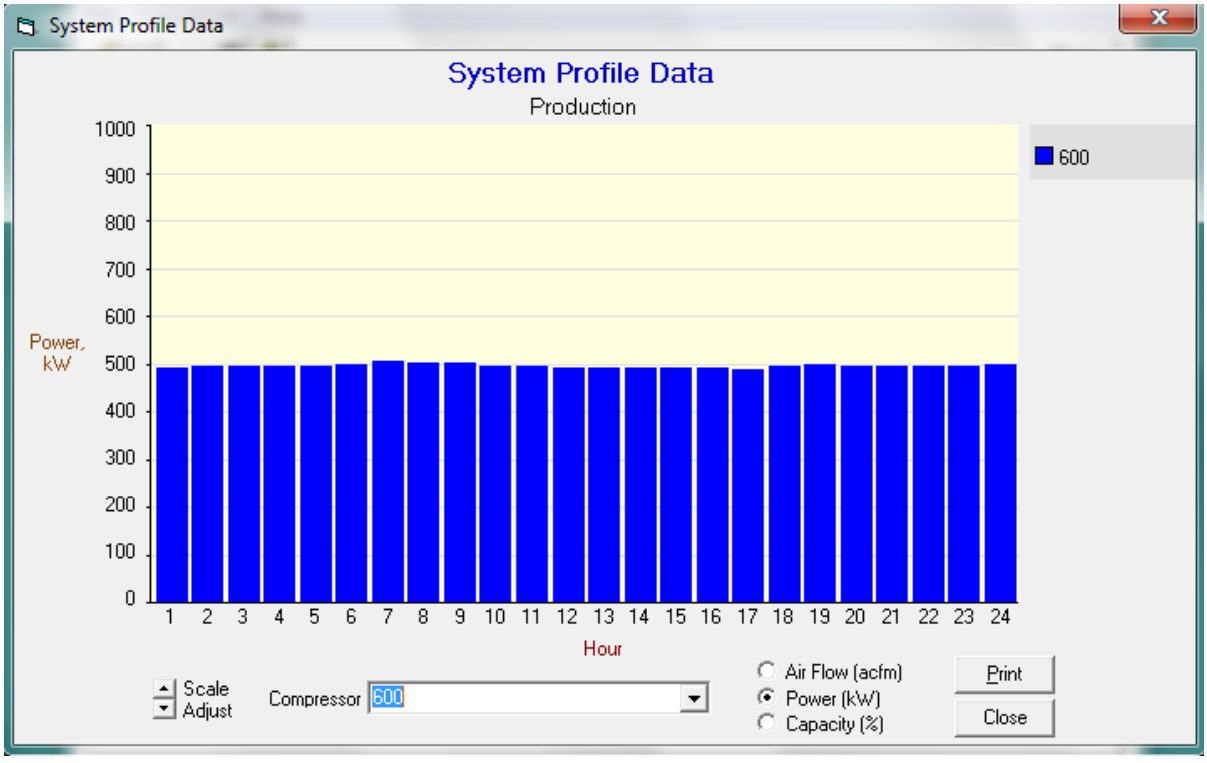


Figure 91: 600 hp Compressor Production Profile (15)

Reducing the system air pressure can result in considerable savings. The current average system air pressure is approximately 103 psig, and it is recommended that the air pressure be dropped to 93 psig, or by roughly 10 psig. Although the average system air pressure is 103 psig, the system pressure is sometimes much higher or lower than 103 psig. For example, pressure rises up to 120 psig when the compressed air demand is low. The compressor power would drop considerably if the average pressure were to be reduced by an average of 10 psig. Reducing the pressure would decrease compressor power because it takes more power to compress air to higher pressure.

After entering the Energy Efficiency Measures module, a new scenario was created to calculate savings by reducing the system pressure, while only considering the 600 hp compressor. The module screen is shown below.

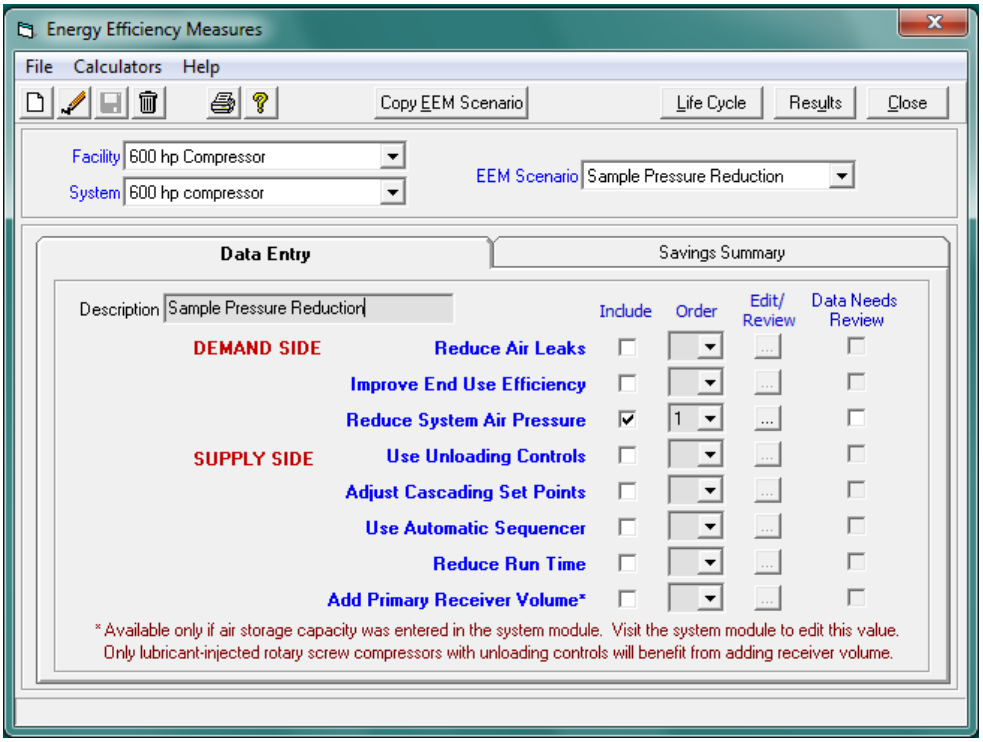


Figure 92: Sample Pressure Reduction Simulation (15)

The results of this measure can be found in the table below.

Table 7: 600 hp System Pressure Reduction Results

Daytype	Total OpHrs	Average Airflow (acfm)	Peak Demand (kW)	Annual Energy (kWh)	Annual Energy Cost	Demand Cost
Baseline	6,000	2,947	505.8	2,971,579	$127,481	$70,587
After Pressure Reduction	6,000	2,796	456.4	2,749,906	$117,971	$64,949

After reducing the pressure, there are reductions in average airflow, demand, and energy. The following table details savings.

Table 8: Savings from Reducing Compressor Pressure

	Total OpHrs	Average Airflow Reduction (acfm)	Peak Demand Reduction (kW)	Annual Energy Savings (kWh)	Annual Energy Cost Savings	Demand Cost Savings	Total Cost Savings
Savings	6,000	151	49.4	221,673	$9,510	$5,638	$15,148

The measure reduces the average air flow by 151 acfm, the peak demand by 49.4 kW, and the yearly energy use by 221,673 kWh. This results in a total cost savings of $15,148/yr. This represents a cost savings of 7.5%. The next step is to validate these findings.

To verify whether the AIRMaster+ calculations are accurate, an alternate method of calculating savings from dropping pressure will now be performed.

Demand Reduction is given by:

$$DR = (1 - FR) \times kW \times \%loaded \div Efficiency \quad \textbf{(Equation 7)}$$

FR is the ratio of proposed power consumption to current power consumption based on operating pressure, no units. FR is based on an analysis using the ideal gas law, and isentropic compression power. The average power of the 600 hp compressor is approximately 500 kW, with an efficiency of 94.5%. The following equation can be used to

estimate the horsepower reduction factor, FR, based on current and proposed operating pressures.

$$FR = \frac{\left(\frac{P_{dp}}{P_i}\right)^{N(k-1)/k} - 1}{\left(\frac{P_{dc}}{P_i}\right)^{N(k-1)/k} - 1} \quad \textbf{(Equation 8)}$$

Where,

P_{dp} = proposed discharge pressure 93 psig, 107.7 psia

P_{dc} = current discharge pressure 103 psig, 117.7 psia

P_i = inlet pressure, 14.7 psia

N = compression factor for two stage screw, 2

k = ratio of specific heat for air, 1.4, no units

The FR ratio will be calculated with a proposed discharge pressure and current discharge pressure of 93 psig, and 103 psig respectively. After entering the values into the above equation, FR is equal to:

FR = 0.9288

Therefore, the demand reduction is:

DR = (1-0.9288) x 500 kW x 100% loaded / 94.5% Efficient

= 37.64 kW

The Energy savings are therefore,

Energy Savings = 37.64 kW x 6,000 hours/yr.

= 225,845 kWh/yr.

The overall cost saving, CS, from the demand and energy reduction are given below:

CS = (225,845 kWh/yr. x $0.043/kWh) + (37.64 kW x $11.63/kW x 12)

= $14,942/yr.

AIRMaster+ gives hour by hour results, in which the pressure may differ from than the average pressure. This is in contrast to calculating savings in the simplistic manner shown above. The formulas offer assumptions as to what the pressure is at all times, and to what pressure the system will be reduced, which also remains static. Despite the inherent differences in complexity, the results are quite similar.

The hand calculations result in overall savings that are $205/yr. less than the results from AIRMaster+, and are about 1.4% less than the AIRMaster+ results. The two calculations are fairly close; accordingly it is reasonable to utilize the AIRMaster+ simulation for forthcoming recommendations.

5.2.2 Automatic Sequencer

Aligning with the main objective of this thesis, an automatic sequencer, which is a form of system master controls, could create a better compressor control system. Controls would allow the compressors to operate more efficiently, thereby saving both energy and money. An automatic sequencer allows the compressed air pressure to be controlled to a lower set point, adhering to a smaller control band. At the same, the sequencer decides, based on pressure and compressed air demand, which compressors are needed or unnecessary. First, reducing the compressed air system pressure will greatly reduce the demand for compressed air; consequently the compressors will see a reduction in power. Secondly, the sequencer might decide to turn off the 600 hp compressor and use the 500 hp compressor with the 100 hp compressor to meet demand. In turn, the operating demand will further reduce, leading to energy and cost savings. Employing an automatic sequencer makes sense for a large compressed air system such as this. AIRMaster+ will simulate the use of an automatic sequencer hour by hour, first reducing the pressure to the desired set point, then choosing which compressors make sense to operate. In addition to operating an automatic sequencer, savings will be shown from decreasing the air leak load in section 5.2.3. For larger compressed air systems, fixing air leaks could realize significant savings.

Entering the Energy Efficiency Measures module in AIRMaster+, the user is given the opportunity to employ the use of an automatic sequencer. It was decided that an appropriate pressure set point would be 95 psig, with a variance of only plus or minus 3 psig. The facility

might have the ability to further decrease the pressure, but this would need to be accomplished in small pressure increments to ensure no production issues occur at the lower pressure. The Energy Efficiency Measures screen is shown in Figure 93 below.

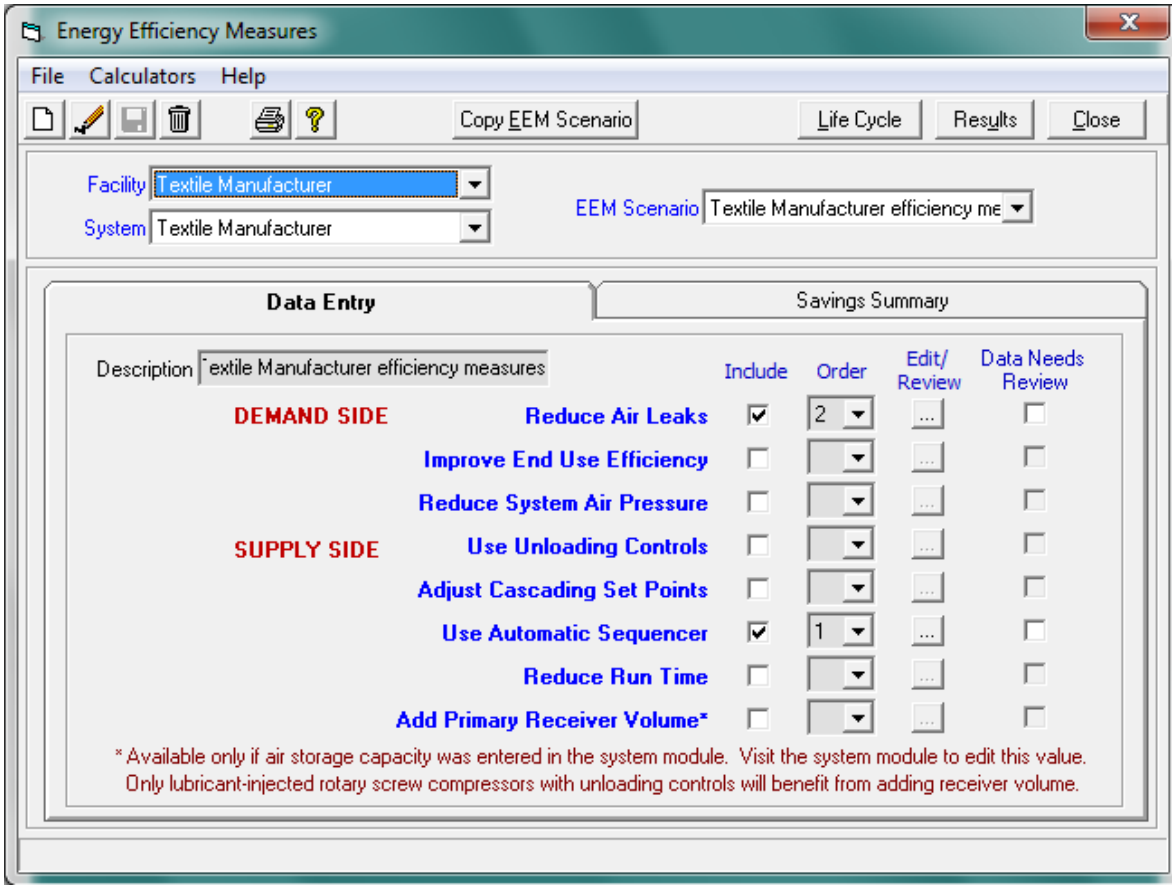


Figure 93: Energy Efficiency Measures (15)

The order of the two measures does matter because the reduced system pressure set point will also diminish the leak load. After clicking the Edit/Review box for the Use Automatic Sequencer, the measure can be prepared to simulate savings. As previously stated, the chosen

pressure set point is 95 psig, with a variance of plus or minus 3 psig. The facility already has a controls system that could have the ability to control with this method; however the system is not being used. Therefore, the implementation of this project could incur a low cost, should it only need to be programmed properly. The cost of a new name brand controls system could be as high as $140,000. Although the actual cost should be quoted, it is estimated that the cost of a new system would be $100,000.

If the current system can be used as a network controls system in the same manner as an automatic sequencer, programming the controls, and training facility personnel to operate the controls, the implementation of this measure would cost much less than a brand new system. The project could incur a low implementation cost should the issue require a simple programming fix. If the issue is deeper, and a decent amount of time is required to resolve any problems, the implementation cost could be as high as $10,000. An estimate of $10,000 will be assumed for simple payback calculations. Figure 94 illustrates the general data tab of the automatic sequencer set-up.

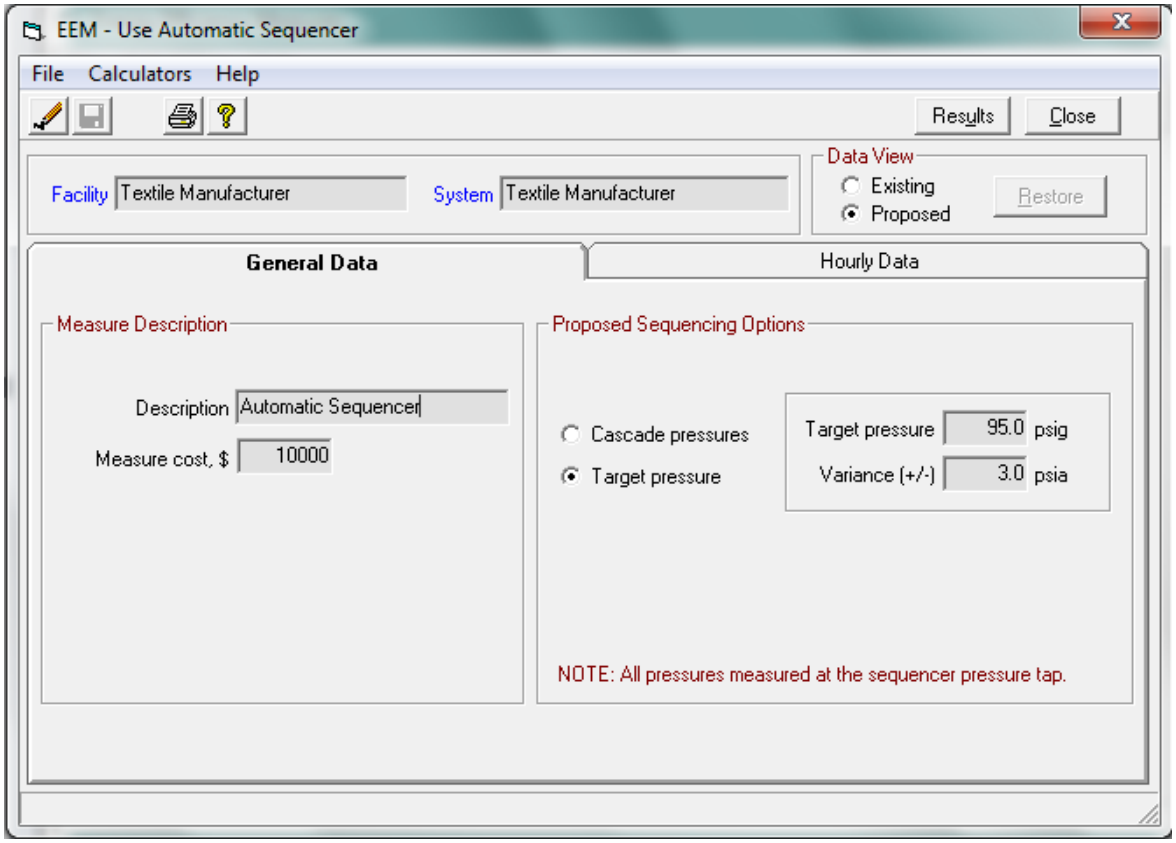


Figure 94: AIRMaster+ Use Automatic Sequencer Measure (15)

It is correspondingly important to properly set up the inputs for each daytype in the hourly data tab. All compressors are made available for each hour of the day, and shut down timers are added to the control. This allows the sequencer to choose the best combination of compressors for operation, and the shut down timers allow the compressors to turn off if they remain unloaded for a period of time. In Appendix A, the complete set-up and savings report will be detailed. Figure 95 shows the hourly data set-up tab.

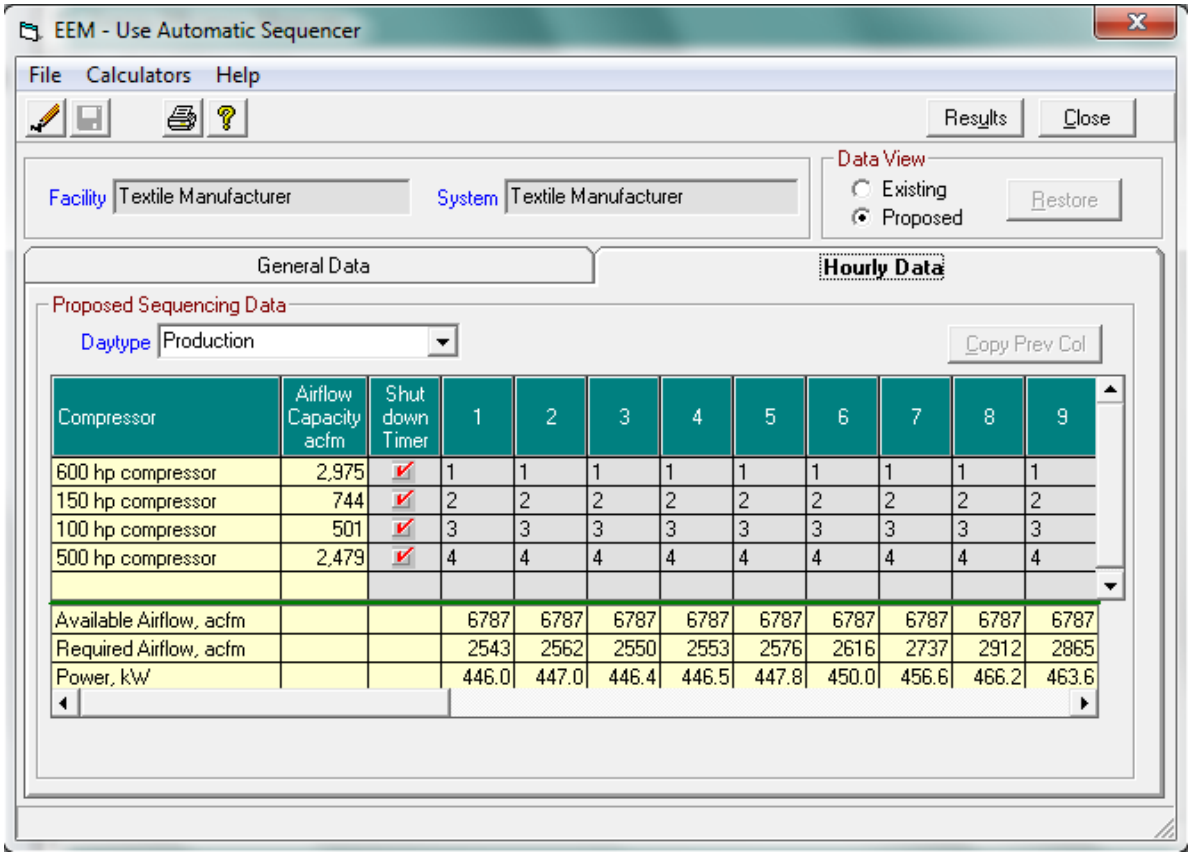

EEM - Use Automatic Sequencer
File Calculators Help
Results Close
Facility Textile Manufacturer
System Textile Manufacturer
Data View
Existing
Proposed
Restore
General Data
Hourly Data
Proposed Sequencing Data
Daytype Production
Copy Prev Col
Compressor | Airflow Capacity acfm | Shut down Timer | 1 | 2 | 3 | 4 | 5 | 6 | 7 | 8 | 9
600 hp compressor | 2,975 | 1 1 1 1 1 1 1 1 1
150 hp compressor | 744 | 2 2 2 2 2 2 2 2 2
100 hp compressor | 501 | 3 3 3 3 3 3 3 3 3
500 hp compressor | 2,479 | 4 4 4 4 4 4 4 4 4
Available Airflow, acfm | 6787 6787 6787 6787 6787 6787 6787 6787 6787
Required Airflow, acfm | 2543 2562 2550 2553 2576 2616 2737 2912 2865
Power, kW | 446.0 447.0 446.4 446.5 447.8 450.0 456.6 466.2 463.6

Figure 95: AIRMaster+ Automatic Sequencer Hourly Data (15)

The measure is now ready for simulation. By exiting to the main Energy Efficiency Measure module screen and pressing the results button on the top right of the screen, the simulation results can be viewed. The baseline results are listed below in Table 9, followed by the automatic sequencer results in Table 10.

Table 9: AIRMaster+ Baseline Operating Results

Daytype	Operating Hours	Average Airflow (acfm)	Peak Demand (kW)	Annual Energy (kWh)	Annual Energy Cost	Demand Cost	Total Cost
Production	6,000	2,857	580.5	3,349,180	$143,680	$81,011	$224,691
Saturday	1,200	1,327	450.8	330,162	$14,164	$0	$14,164
Sunday	350	1,119	203.2	70,507	$3,025	$0	$3,025
Total	7,550	2,553	580.5	3,749,849	$160,869	$81,011	$241,880

Table 10: AIRMaster+ Use Automatic Sequencer Results

Daytype	Operating Hours	Average Airflow (acfm)	Peak Demand (kW)	Annual Energy (kWh)	Annual Energy Cost	Demand Cost	Total Cost
Production	6,000	2,681	466.2	2,721,122	$116,736	$65,062	$181,798
Saturday	1,200	1,246	400.5	291,580	$12,509	$0	$12,509
Sunday	350	1,050	179.0	61,709	$2,647	$0	$2,647
Total	7,550	2,377	467.0	3,074,411	$131,892	$65,062	$196,954

According to the system baseline simulation, the compressed air system uses 3,749,849 kWh/yr., resulting in a total of $241,880/yr. in operating costs. The introduction of the automatic sequencer results in an annual energy consumption of 3,074,665 kWh, which yields a total of $196,954 in yearly operating costs. The savings are shown in the Table 11 below.

Table 11: AIRMaster+ Savings Resulting from Automatic Sequencer

Daytype	Operating Hours	Average Airflow Reduction (acfm)	Peak Demand Reduction (kW)	Annual Energy Savings (kWh)	Annual Energy Cost Savings	Demand Cost Savings	Total Cost Savings
Production	6,000	176	114	628,058	26,944	15,949	42,893
Saturday	1,200	81	50	38,582	1,655	0	1,655
Sunday	350	69	24	8,798	377	0	377
Total	7,550	176	114.0	675,438	$28,976	$15,949	$44,925

The implementation of an automatic sequencer results in substantial savings. The energy savings is 675,438 kWh/yr., and the peak demand reduction is 114 kW. This results in a total annual operation cost savings of $44,795, representing a savings of 18%.

5.2.3 Reduce Air Leaks

By reducing the compressed air volumetric flow through leaks in the compressed air system, the compressors will consume less power and energy. Air leaks cause unnecessary compressed air demand, which causes the compressor to run longer or at a higher power. The compressed air volumetric rate associated with the leak depends on the system air pressure and the size of the leak. The formula to calculate volumetric flow of free air associated with an air leak is given below:

$$Vf = \frac{C_1 \times P_0 \times C_d \times D^2 \times (T_i + 460)}{C_2 \times P_i \times \sqrt{T_0 + 460}} \quad \textbf{(Equation 9)}$$

Where,

V_f = volumetric flow of free air, CFM

C_1 = choked flow constant, 1336 ft/min·°$R^{0.5}$

P_o = compressor operating pressure

C_d = discharge coefficient for orifice, 0.6, no units

D = leak diameter, inches

T_i = inlet temperature

C_2 = conversion constant, 144 in^2/ft

P_i = inlet (atmospheric) pressure, 14.7 psia

T_o = average line temperature

The compressor power required to supply the compressed air for each leak is given in the formula below.

$$Power\ \mathrm{Re}quired = \frac{P_i \times C_2 \times V_f \times \frac{k}{k-1} \times N \times C_3 \times \left[\left(\frac{P_o}{\{P_i} \right)^{\frac{k-1}{k \times N}} - 1 \right]}{E_v \times E_m}$$

where,

L = power loss, hp

C_2 = conversion constant, 144 in^2/ft^2

k = specific heat ratio of air, 1.4, no units

N = factor based on type of compressor considered,

1.25 for single stage flooded oil screw; 2 for two stage

C_3 = conversion constant, 3.03 x 10^{-5} hp·min/ft·lb

E_v = air compressor volumetric efficiency, 85%, no units

E_m = compressor motor efficiency

Because of the way typical compressors operate, even if all leaks are eliminated, the entirety of the power required to satisfy those leaks will not be saved. Typically, a modulating compressor still uses 65 to 70 percent of its full load power at zero percent capacity. Thus, only 30 to 35% of that power required to supply the air leaks will be realized in savings.

AIRMaster+ will calculate savings from the elimination of a designated volumetric flow of compressed air. Typically, a facility can expect 10 percent of compressor capacity to be leak related. AIRMaster+ requires that the user specify the percent capacity of a compressor that is dedicated to supplying leaks. As the 100 hp compressor is the trim compressor, any reduction in compressed air leaks will affect its percent capacity and power. It was approximated that 50% of the 100 hp capacity is given to leaks that can be eliminated, resulting in 251 CFM of compressed air blowing to atmosphere. The Reduce Airleaks set-up is shown in Figure 96 below.

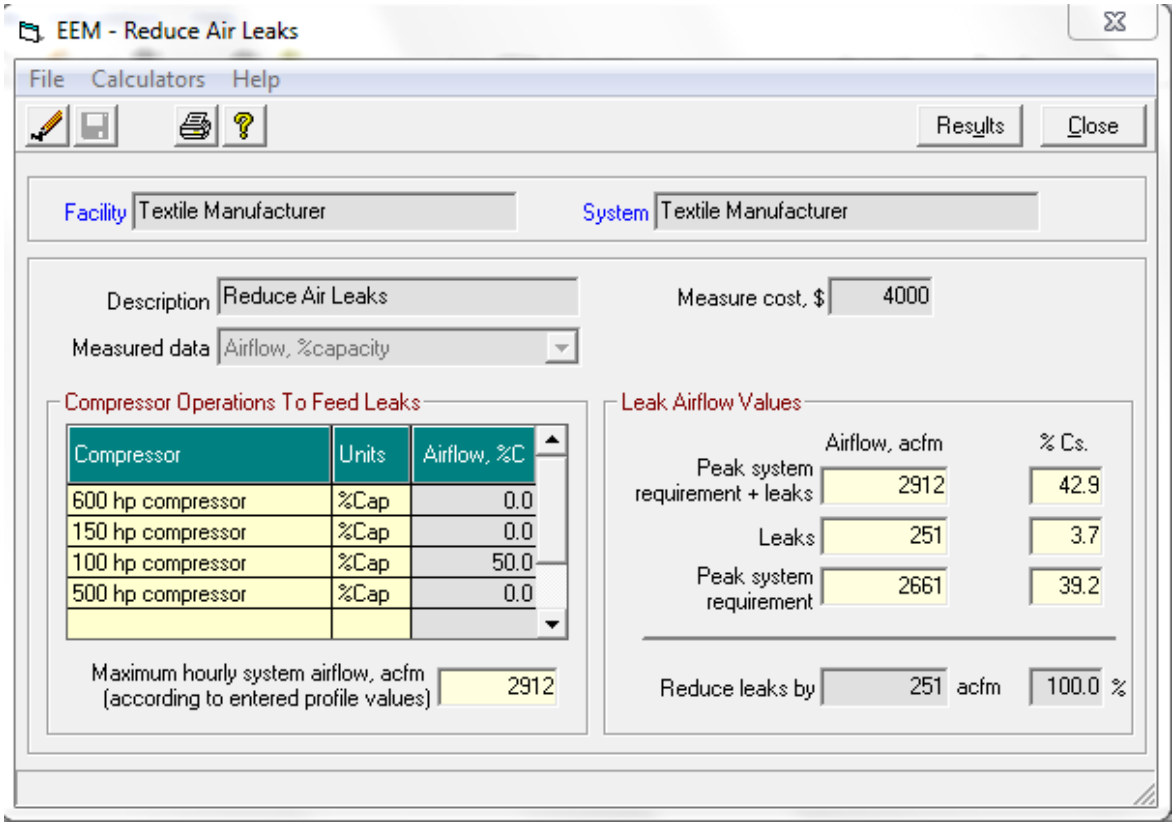


Figure 96: AIRMaster+ Reduce Air Leaks

After simulating this scenario, AIRMaster+ calculated a new annual energy usage, and total electricity cost. The new annual energy usage is 2,945,413 kWh, which is a reduction of 128,998 kWh per year. The results from reducing air leaks are listed in Table 12

Table 12: AIRMaster+ Reduce Air Leaks Results

Daytype	Operating Hours	Average Airflow (acfm)	Peak Demand (kW)	Annual Energy (kWh)	Annual Energy Cost	Demand Cost	Total Cost
Production	6,000	2,430	452.5	2,638,725	$113,201	$63,145	$176,346
Saturday	1,200	995	386.7	258,320	$11,082	$0	$11,082
Sunday	350	800	144.6	48,367	$2,075	$0	$2,075
Total	7,550	2,126	452.5	2,945,412	$126,358	$63,145	$189,503

The savings from reducing air leaks are detailed in Table 13. The total annual cost savings is $7,451. This represents a savings of 3.4% of the operational costs.

Table 13: AIRMaster+Reduce Air Leaks Savings

Daytype	Operating Hours	Average Airflow Reduction (acfm)	Peak Demand Reduction (kW)	Annual Energy Savings (kWh)	Annual Energy Cost Savings	Demand Cost Savings	Total Cost Savings
Production	6,000	251	14	82,397	3,535	1,917	5,452
Saturday	1,200	251	14	33,260	1,427	0	1,427
Sunday	350	250	34	13,342	572	0	572
Total	7,550	251	14.0	128,999	$5,534	$1,917	$7,451

5.2.4 Total Savings

The two measures result in a total annual savings of $52,377, representing 21.4% reduction in compressor electrical costs. It should be noted that the actual savings could be lower or higher than the simulated savings. This model represents the best estimation of savings

resulting from the implementation of these two measures. The total savings summary is detailed below.

Table 14: AIRMaster+ Savings Summary

Description	Energy Savings (kWh)	Annual Energy Cost Savings	Demand Savings	Demand Cost Savings	Total Savings	Implement Cost	Simple Payback (months)
Automatic Sequencer	675,438	$28,976	114.3	$15,950	$44,926	$10,000	3
Reduce Air Leaks	128,998	$5,534	13.7	$1,917	$7,451	$4,000	7
Total	804,436	$34,510		$17,867	$52,377	$14,000	4

Chapter 6 – Conclusions

6.1 Data Analysis Conclusions

An important analysis performed in this study was determining a method to calculate power factor, which is an important factor when calculating compressor power. When a compressor is operating around full load amperage the power factor is likely between 0.85 and 0.9, but the power factor can be much lower when the compressor is operating at part load. For example, when a load/unload flooded oil rotary screw compressor is fully unloaded, it may have a power factor as low as 0.3. Conversely, at full load amperage, the same compressor might have a power factor of 0.87. Consequently, power factor greatly affects the actual compressor power, and therefore is important to have a means to quickly estimate the power factor.

AIRMaster+ has a compressor power calculator, which it uses during its simulations. Using the 100 hp compressor, the power was calculated with input amperages from 0 to its full load amperage of 135 amps. The calculated compressor power was divided by the power at the same amperage assuming a power factor of one, resulting in the AIRMaster+ calculated power factor. The calculated power factor was then plotted against the calculated compressor power, which was then used to perform a 6^{th} order polynomial regression. The resulting polynomial equation could be used to determine the power factor at a specific compressor power reading. This first trendline equation is only useful for the 100 hp compressor.

Therefore, to form a power factor equation that is applicable to all compressors, the input amperage was divided by the full load amperage to give the percent full load amperage. Then a plot of power factor against percent full load amperage was made, resulting in a 6^{th} order polynomial trendline equation. This equation is particularly useful, as the power factor can now be calculated for any sized motor based on the percent full load amperage. Furthermore, this equation can be utilized during any future compressor power analyses.

It can be quite worthwhile to collect data about the compressed air demand. Knowing information about a facilities demand for compressed air can help determine periods of the production day, in which compressed air demand is high. Volumetric flow can also help determine inefficiencies in the operation of the compressed air system in relation to system pressure. For example, during the transition from normal compressed air demand periods to low demand periods, one would expect lower compressor capacity and the pressure to remain relatively static. One can find issue with the way the compressed air system is operating if the pressure rises significantly above the desired setpoint. Additionally, during the transition to high compressed air demand periods, one can expect the compressor capacity to increase until demand is met, and the pressure will likely decrease to below setpoint.

By knowing more about the operating parameters of the compressed air system, one can begin to make better judgments about the system. However, it is not always possible to collect all of the data necessary, for instance volumetric flow. For this reason, it could be

greatly important to accurately estimate a volumetric flow for a given compressed air system, which was necessary during this study. During the first data collection period, volumetric flow was not obtainable, but was for the second data collection period. Likewise, pressure data was collected during the first data collection period, but due to data logger error, pressure was not collected during the second period of data collection. Therefore, it was required to estimate the compressor capacity for the first set of data. Using data from the second data collection, a linear regression was performed on a plot of volumetric flow against compressor power. Through this analysis, it was determined that there is a strong linear correlation between compressor power and volumetric flow, and that volumetric flow in cubic feet per minute is equal to approximately 4.44 multiplied by the compressor horsepower .

This estimation is quite accurate at full load power, but during facility shut down, start-up, and periods of part load operation, the estimation lacks accuracy. The compressor system is fully loaded approximately 97% of the time, and therefore the volumetric flow estimation is only about 3% different than the actual value 97% of the time. The estimation of 4.44 CFM per horsepower is likely translatable to other compressed air systems, and verification of this could be an area of future work.

Additionally, observations were made regarding the relationship between pressure and compressor power. It was shown that pressure is difficult to predict based on compressor

power. This is because pressure can decrease when compressor power increases, increase when power increases, decrease when power decreases, and finally pressure can increase when power decreases. As seen in Figure 59, in section 4.3.8, pressure can decrease as the compressor power increases. This is a typical and expected trend associated with using modulating compressors.as compressed air demand increases. At low demand, pressure is high and the compressors can operate at a higher modulating percentage. As the compressed air demand increases, the butterfly valve begins to open up and the compressor power increases. The pressure continues to decrease until the compressors are at full capacity and the demand is met. This indicates a negative correlation between pressure and brake horsepower. Ultimately, observations such as these help to illustrate compressor operation, and help determine any potential issues with the subject compressed air system.

6.2 AIRMaster+ Conclusions

As a result of data collection, analysis, and modeling with the AIRMaster+, the operation cost of the facility's compressed air system can be greatly reduced by the implementation of network controls, and the possible reduction of air leaks. The automatic sequencer will first act to reduce the system air pressure to 95 psig, with a variance of 3 psig. This would control the system pressure within 92 psig and 98 psig. Secondly the sequencer would determine which compressors are to operate based on compressed air demand. During data collection, all of the compressors turned on during operation start-up, which is not necessary and could set a high billing demand. An automatic sequencer would avoid this type of operation.

After modeling the facility's compressed air system in AIRMaster+, the first energy efficiency measure to be simulated was the implementation of the automatic sequencer. This measure offers an energy savings of 675,438 kWh annually. Including demand reduction cost savings, the total savings from this simulation is $44,926 annually. The implementation of this measure should require minimal involvement. The facility already owns a control system, thus the cost will likely only originate from reprogramming, which is estimated to cost no more than $10,000.

The second efficiency measure that AIRMaster+ simulated is the reduction in air leaks. This measure requires finding and fixing compressed air leaks as they occur, which will reduce compressor power. The recommendation offers an energy savings of 128,998 kWh per year, resulting in a total operational cost savings $7,451 per year.

The combination of the implementation of an automatic sequencer, and the reduction of air leaks results in a reduction of 804,436 kWh per year, and a total annual savings of $52,377, representing 21.4% reduction in compressor electrical costs.

Ensuring that the compressor controls are operating properly will help to improve the compressor system operation. The system pressure will be reduced and controlled to a tighter pressure band, both reducing compressor power and improving overall system operation. The compressors will have the capability to quickly respond to demand events and avoid

operation leading to inefficiencies such as pressure run-up. Additionally, while reducing air leaks is not a complicated measure, any reduction in operation cost will directly influence the facility profit.

6.3 Further Opportunities

While this project is associated primarily with improving the overall compressor system controls, part load control is an area of possible improvement. From data analysis, there is often a smaller compressor that is partly loaded, using a high percentage of full load power at lower compressor capacities. If the facility were to purchase a variable speed drive for one of the smaller compressors, the part load efficiency would increase. A variable speed drive has approximately a one to one ratio between percent compressor capacity and percent full load compressor power. Hence, if the variable speed drive compressor is at 50 percent capacity, the compressor will use 50 percent of its full load power. A compressor using load/unload or modulating controls cannot achieve this ratio, and therefore are more expensive to operate at part load. While this opportunity could offer significant energy and cost savings, the implementation of a variable speed drive is quite costly, therefore only a complete analysis could determine the practicality of such a project.

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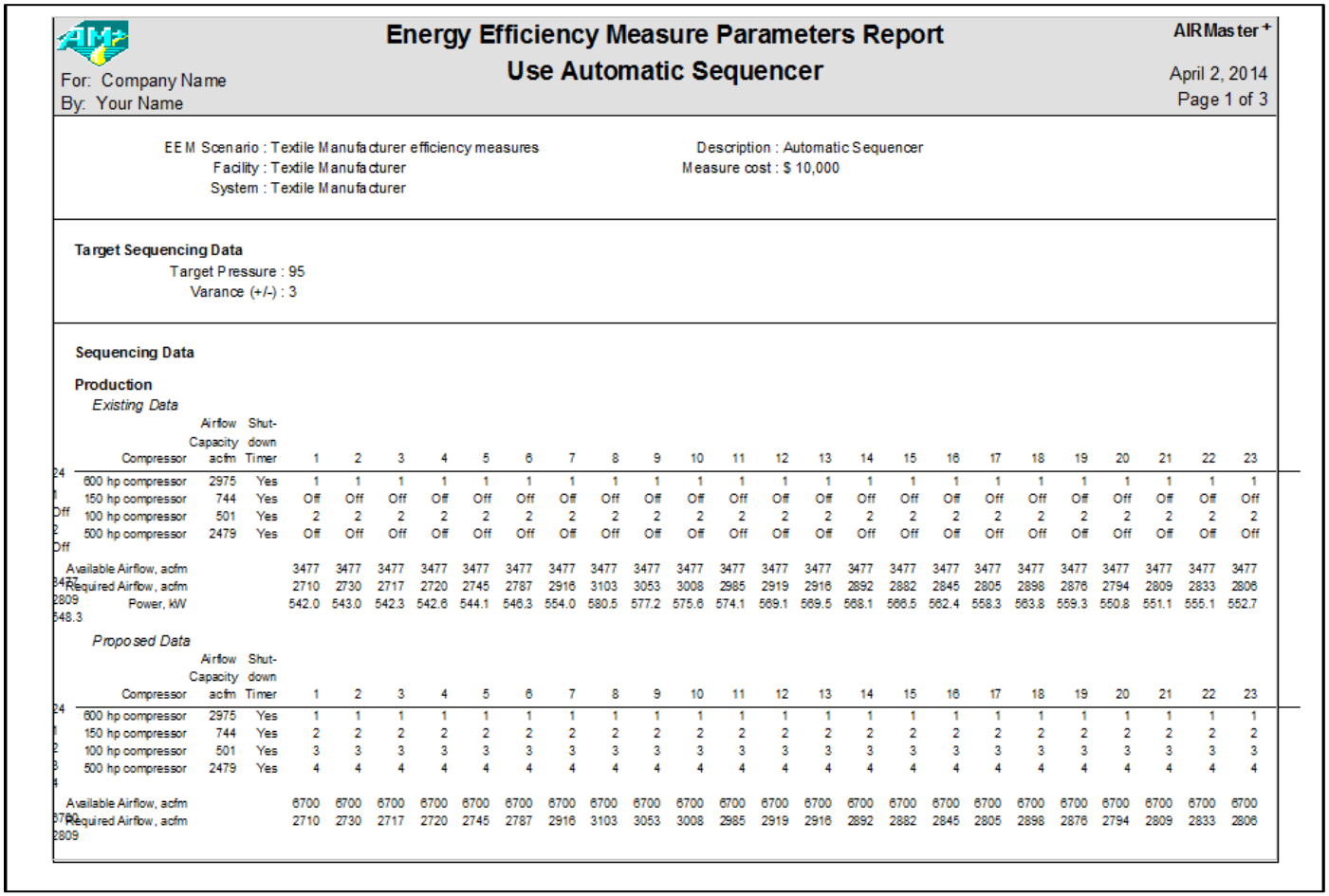
APPENDICES

Appendix A-AIRMaster+ Automatic Sequencer Existing and Proposed Scenarios



For: Company Name
By: Your Name

Energy Efficiency Measure Parameters Report
Use Automatic Sequencer

AIRMaster+
April 2, 2014
Page 1 of 3

EEM Scenario : Textile Manufacturer efficiency measures
Facility : Textile Manufacturer
System : Textile Manufacturer

Description : Automatic Sequencer
Measure cost : $ 10,000

Target Sequencing Data

Target Pressure : 95
Varance (+/-) : 3

Sequencing Data

Production

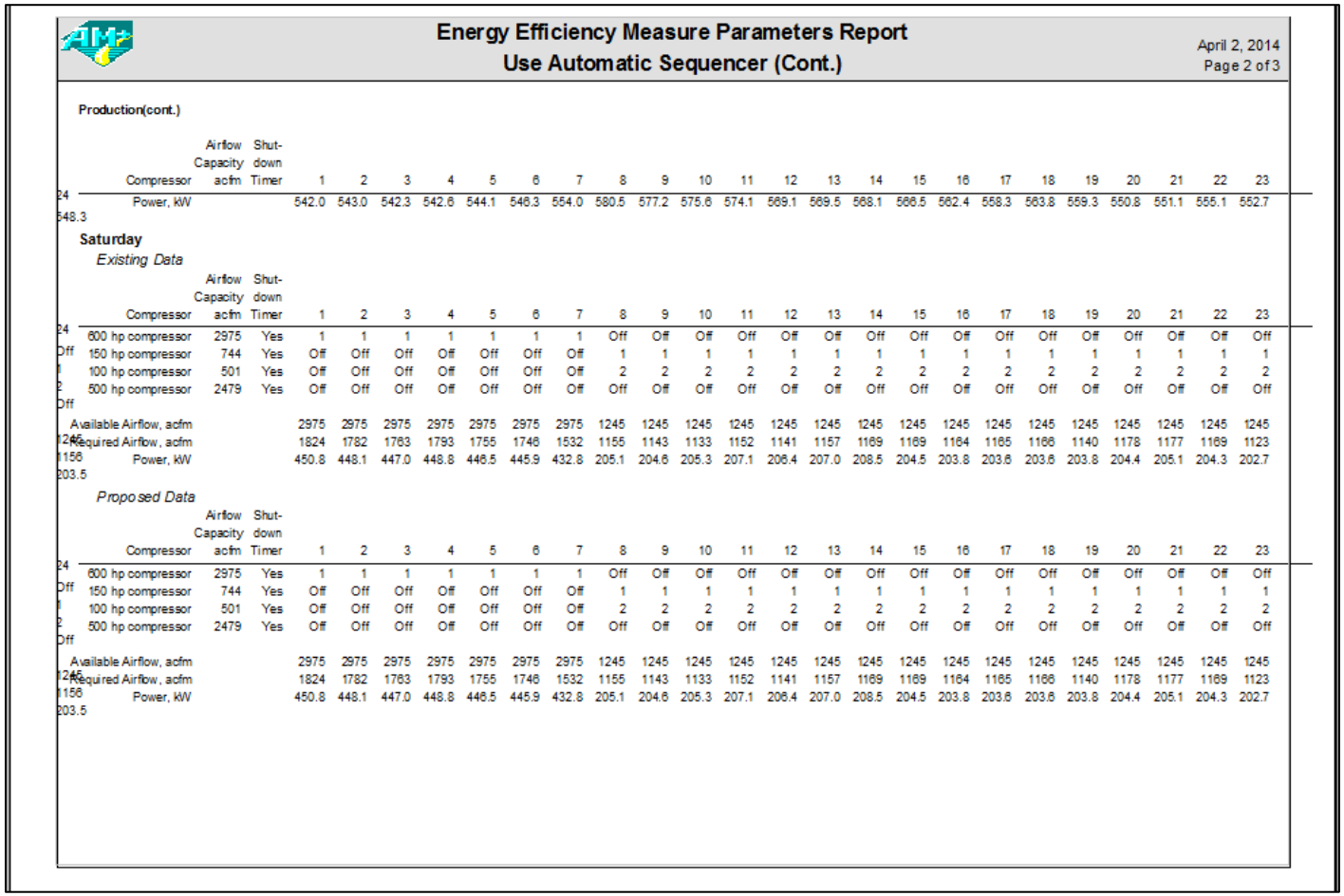
Existing Data

Compressor	Airflow Capacity acfm	Shut-down Timer	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23
600 hp compressor	2975	Yes	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1
150 hp compressor	744	Yes	Off	Off	Off	Off	Off	Off	Off	Off	Off	Off	Off	Off	Off	Off	Off	Off	Off	Off	Off	Off	Off	Off	Off
100 hp compressor	501	Yes	2	2	2	2	2	2	2	2	2	2	2	2	2	2	2	2	2	2	2	2	2	2	2
500 hp compressor	2479	Yes	Off	Off	Off	Off	Off	Off	Off	Off	Off	Off	Off	Off	Off	Off	Off	Off	Off	Off	Off	Off	Off	Off	Off
Available Airflow, acfm			3477	3477	3477	3477	3477	3477	3477	3477	3477	3477	3477	3477	3477	3477	3477	3477	3477	3477	3477	3477	3477	3477	3477
Required Airflow, acfm			2710	2730	2717	2720	2745	2787	2916	3103	3053	3008	2985	2919	2916	2892	2882	2845	2805	2898	2876	2794	2809	2833	2806
Power, kW			542.0	543.0	542.3	542.6	544.1	546.3	554.0	580.5	577.2	575.6	574.1	569.1	569.5	568.1	566.5	562.4	558.3	563.8	559.3	550.8	551.1	555.1	552.7

Proposed Data

Compressor	Airflow Capacity acfm	Shut-down Timer	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23
600 hp compressor	2975	Yes	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1
150 hp compressor	744	Yes	2	2	2	2	2	2	2	2	2	2	2	2	2	2	2	2	2	2	2	2	2	2	2
100 hp compressor	501	Yes	3	3	3	3	3	3	3	3	3	3	3	3	3	3	3	3	3	3	3	3	3	3	3
500 hp compressor	2479	Yes	4	4	4	4	4	4	4	4	4	4	4	4	4	4	4	4	4	4	4	4	4	4	4
Available Airflow, acfm			6700	6700	6700	6700	6700	6700	6700	6700	6700	6700	6700	6700	6700	6700	6700	6700	6700	6700	6700	6700	6700	6700	6700
Required Airflow, acfm			2710	2730	2717	2720	2745	2787	2916	3103	3053	3008	2985	2919	2916	2892	2882	2845	2805	2898	2876	2794	2809	2833	2806

A 1: Automatic Sequencer Production Existing and Proposed Data



Energy Efficiency Measure Parameters Report
Use Automatic Sequencer (Cont.)

April 2, 2014
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Production(cont.)

Compressor	Airflow Capacity acfm	Shut-down Timer	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23	24
Power, kW			542.0	543.0	542.3	542.6	544.1	546.3	554.0	580.5	577.2	575.6	574.1	569.1	569.5	568.1	566.5	562.4	558.3	563.8	559.3	550.8	551.1	555.1	552.7	548.3

Saturday

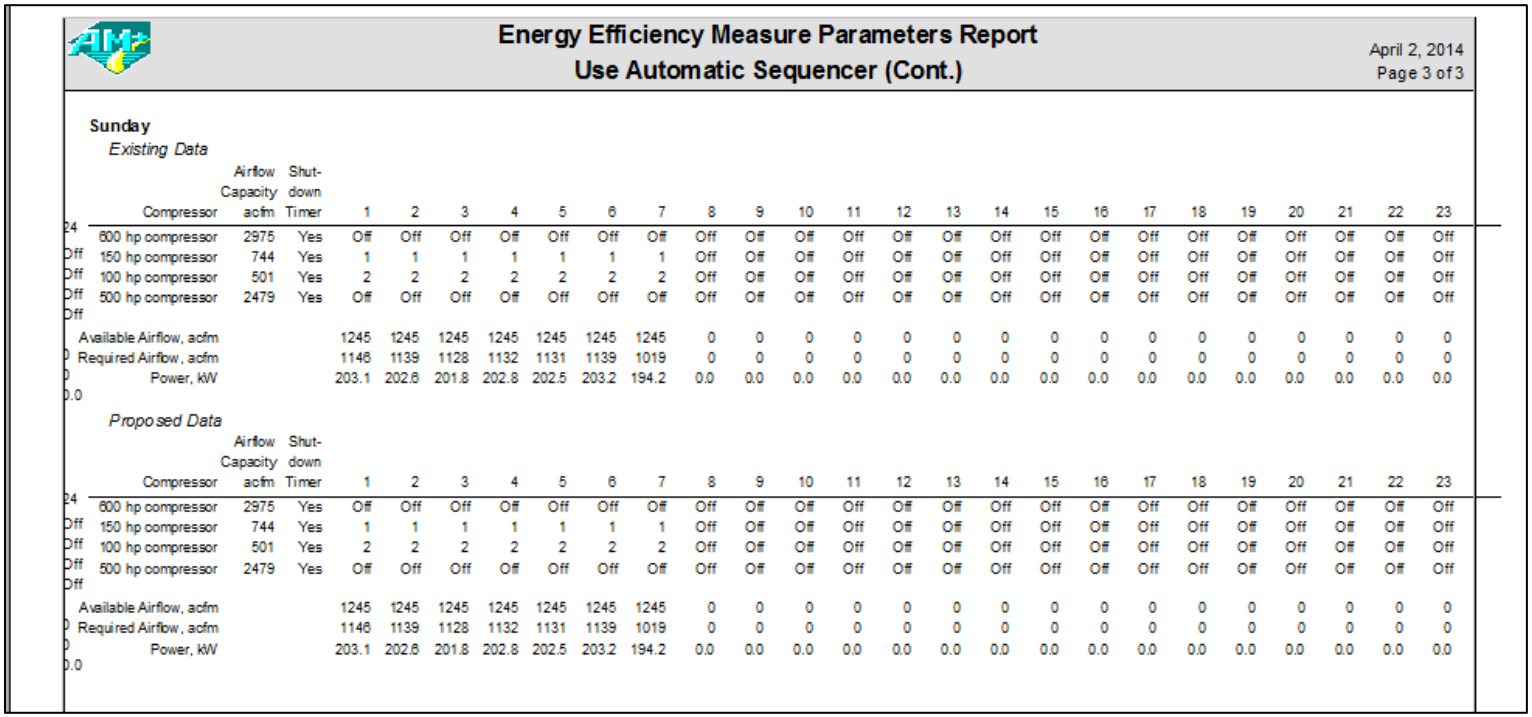
Existing Data

Compressor	Airflow Capacity acfm	Shut-down Timer	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23	24
600 hp compressor	2975	Yes	1	1	1	1	1	1	1	Off	Off	Off	Off	Off	Off	Off	Off	Off	Off	Off	Off	Off	Off	Off	Off	Off
150 hp compressor	744	Yes	Off	Off	Off	Off	Off	Off	Off	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	Off
100 hp compressor	501	Yes	Off	Off	Off	Off	Off	Off	Off	2	2	2	2	2	2	2	2	2	2	2	2	2	2	2	2	Off
500 hp compressor	2479	Yes	Off	Off	Off	Off	Off	Off	Off	Off	Off	Off	Off	Off	Off	Off	Off	Off	Off	Off	Off	Off	Off	Off	Off	Off
Available Airflow, acfm			2975	2975	2975	2975	2975	2975	2975	1245	1245	1245	1245	1245	1245	1245	1245	1245	1245	1245	1245	1245	1245	1245	1245	1245
Required Airflow, acfm			1824	1782	1763	1793	1755	1746	1532	1155	1143	1133	1152	1141	1157	1169	1169	1164	1165	1166	1140	1178	1177	1169	1123	1156
Power, kW			450.8	448.1	447.0	448.8	446.5	445.9	432.8	205.1	204.6	205.3	207.1	206.4	207.0	208.5	204.5	203.8	203.6	203.6	203.8	204.4	205.1	204.3	202.7	203.5

Proposed Data

Compressor	Airflow Capacity acfm	Shut-down Timer	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23	24
600 hp compressor	2975	Yes	1	1	1	1	1	1	1	Off	Off	Off	Off	Off	Off	Off	Off	Off	Off	Off	Off	Off	Off	Off	Off	Off
150 hp compressor	744	Yes	Off	Off	Off	Off	Off	Off	Off	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	Off
100 hp compressor	501	Yes	Off	Off	Off	Off	Off	Off	Off	2	2	2	2	2	2	2	2	2	2	2	2	2	2	2	2	Off
500 hp compressor	2479	Yes	Off	Off	Off	Off	Off	Off	Off	Off	Off	Off	Off	Off	Off	Off	Off	Off	Off	Off	Off	Off	Off	Off	Off	Off
Available Airflow, acfm			2975	2975	2975	2975	2975	2975	2975	1245	1245	1245	1245	1245	1245	1245	1245	1245	1245	1245	1245	1245	1245	1245	1245	1245
Required Airflow, acfm			1824	1782	1763	1793	1755	1746	1532	1155	1143	1133	1152	1141	1157	1169	1169	1164	1165	1166	1140	1178	1177	1169	1123	1156
Power, kW			450.8	448.1	447.0	448.8	446.5	445.9	432.8	205.1	204.6	205.3	207.1	206.4	207.0	208.5	204.5	203.8	203.6	203.6	203.8	204.4	205.1	204.3	202.7	203.5

A 2: Automatic Sequencer Saturday Existing and Proposed Data



Energy Efficiency Measure Parameters Report
Use Automatic Sequencer (Cont.)

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Sunday

Existing Data

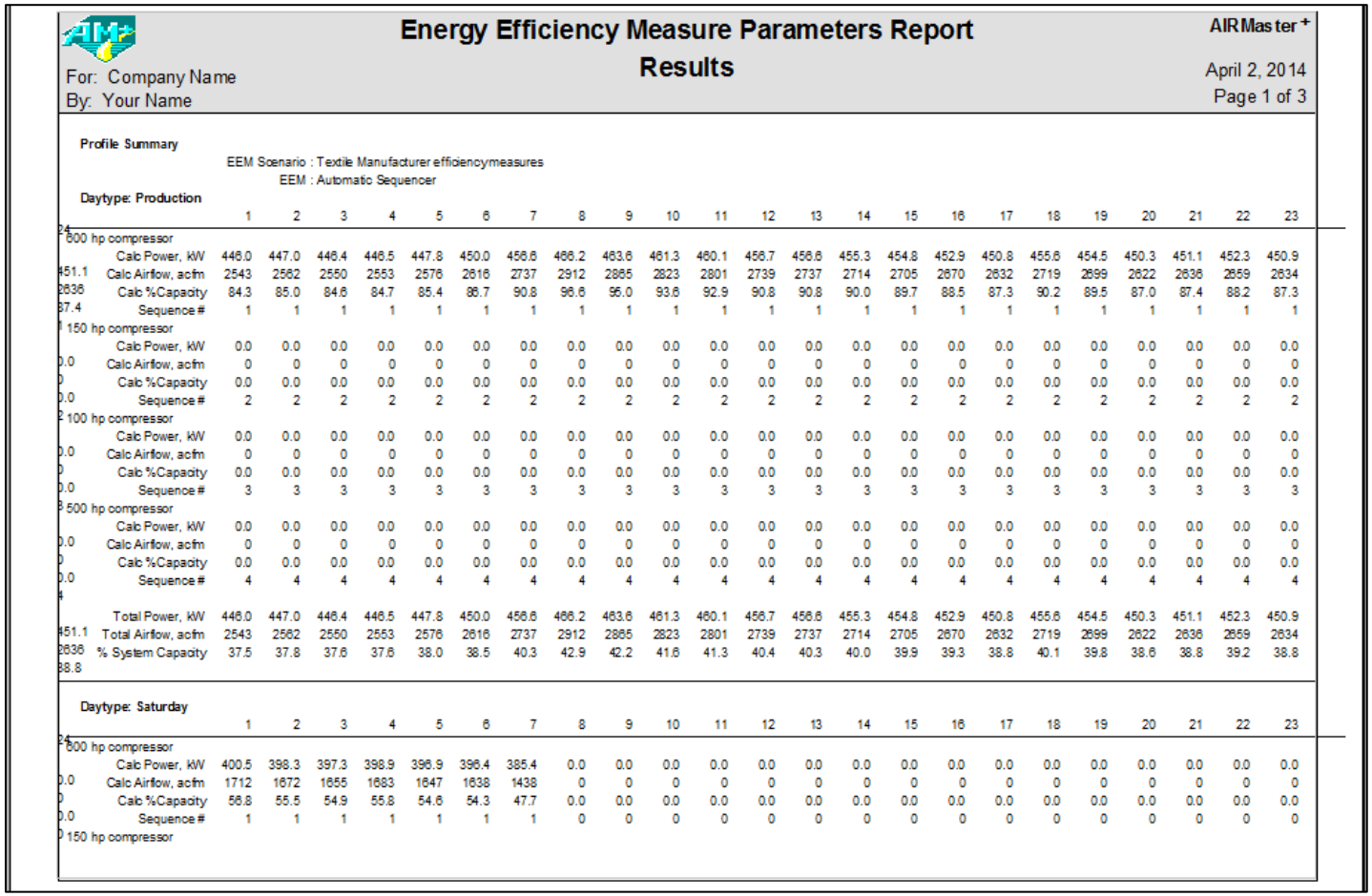
Compressor	Airflow Capacity acfm	Shut-down Timer	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23
600 hp compressor	2975	Yes	Off	Off	Off	Off	Off	Off	Off	Off	Off	Off	Off	Off	Off	Off	Off	Off	Off	Off	Off	Off	Off	Off	Off
150 hp compressor	744	Yes	1	1	1	1	1	1	1	Off	Off	Off	Off	Off	Off	Off	Off	Off	Off	Off	Off	Off	Off	Off	Off
100 hp compressor	501	Yes	2	2	2	2	2	2	2	Off	Off	Off	Off	Off	Off	Off	Off	Off	Off	Off	Off	Off	Off	Off	Off
500 hp compressor	2479	Yes	Off	Off	Off	Off	Off	Off	Off	Off	Off	Off	Off	Off	Off	Off	Off	Off	Off	Off	Off	Off	Off	Off	Off
Available Airflow, acfm			1245	1245	1245	1245	1245	1245	1245	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
Required Airflow, acfm			1146	1139	1128	1132	1131	1139	1019	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
Power, kW			203.1	202.6	201.8	202.8	202.5	203.2	194.2	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0

Proposed Data

Compressor	Airflow Capacity acfm	Shut-down Timer	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23
600 hp compressor	2975	Yes	Off	Off	Off	Off	Off	Off	Off	Off	Off	Off	Off	Off	Off	Off	Off	Off	Off	Off	Off	Off	Off	Off	Off
150 hp compressor	744	Yes	1	1	1	1	1	1	1	Off	Off	Off	Off	Off	Off	Off	Off	Off	Off	Off	Off	Off	Off	Off	Off
100 hp compressor	501	Yes	2	2	2	2	2	2	2	Off	Off	Off	Off	Off	Off	Off	Off	Off	Off	Off	Off	Off	Off	Off	Off
500 hp compressor	2479	Yes	Off	Off	Off	Off	Off	Off	Off	Off	Off	Off	Off	Off	Off	Off	Off	Off	Off	Off	Off	Off	Off	Off	Off
Available Airflow, acfm			1245	1245	1245	1245	1245	1245	1245	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
Required Airflow, acfm			1146	1139	1128	1132	1131	1139	1019	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
Power, kW			203.1	202.6	201.8	202.8	202.5	203.2	194.2	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0

A 3: Automatic Sequencer Sunday Existing and Proposed Data

Appendix B-AIRMaster+ Automatic Sequencer Results



Energy Efficiency Measure Parameters Report
Results

AIRMaster+
April 2, 2014
Page 1 of 3

For: Company Name
By: Your Name

Profile Summary

EEM Scenario : Textile Manufacturer efficiencymeasures
EEM : Automatic Sequencer

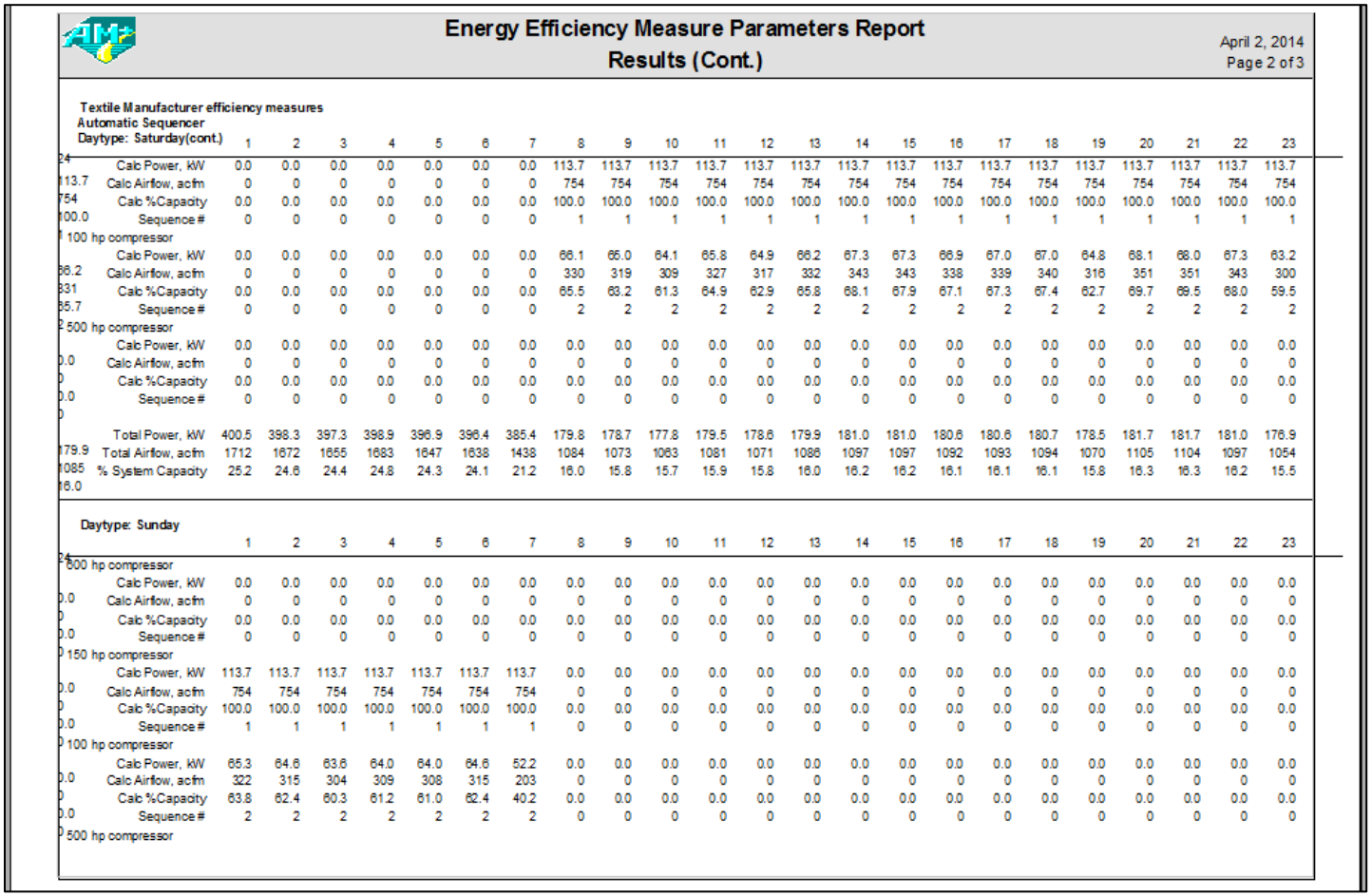
Daytype: Production

	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23
600 hp compressor																							
Calc Power, kW	446.0	447.0	446.4	446.5	447.8	450.0	456.6	466.2	463.6	461.3	460.1	456.7	456.6	455.3	454.8	452.9	450.8	455.6	454.5	450.3	451.1	452.3	450.9
Calc Airflow, acfm	2543	2562	2550	2553	2576	2616	2737	2912	2865	2823	2801	2739	2737	2714	2705	2670	2632	2719	2699	2622	2636	2659	2634
Calc %Capacity	84.3	85.0	84.6	84.7	85.4	86.7	90.8	96.6	95.0	93.6	92.9	90.8	90.8	90.0	89.7	88.5	87.3	90.2	89.5	87.0	87.4	88.2	87.3
Sequence #	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1
150 hp compressor																							
Calc Power, kW	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
Calc Airflow, acfm	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
Calc %Capacity	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
Sequence #	2	2	2	2	2	2	2	2	2	2	2	2	2	2	2	2	2	2	2	2	2	2	2
100 hp compressor																							
Calc Power, kW	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
Calc Airflow, acfm	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
Calc %Capacity	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
Sequence #	3	3	3	3	3	3	3	3	3	3	3	3	3	3	3	3	3	3	3	3	3	3	3
500 hp compressor																							
Calc Power, kW	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
Calc Airflow, acfm	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
Calc %Capacity	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
Sequence #	4	4	4	4	4	4	4	4	4	4	4	4	4	4	4	4	4	4	4	4	4	4	4
Total Power, kW	446.0	447.0	446.4	446.5	447.8	450.0	456.6	466.2	463.6	461.3	460.1	456.7	456.6	455.3	454.8	452.9	450.8	455.6	454.5	450.3	451.1	452.3	450.9
Total Airflow, acfm	2543	2562	2550	2553	2576	2616	2737	2912	2865	2823	2801	2739	2737	2714	2705	2670	2632	2719	2699	2622	2636	2659	2634
% System Capacity	37.5	37.8	37.6	37.6	38.0	38.5	40.3	42.9	42.2	41.6	41.3	40.4	40.3	40.0	39.9	39.3	38.8	40.1	39.8	38.6	38.8	39.2	38.8

Daytype: Saturday

	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23
600 hp compressor																							
Calc Power, kW	400.5	398.3	397.3	398.9	396.9	396.4	385.4	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
Calc Airflow, acfm	1712	1672	1655	1683	1647	1638	1438	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
Calc %Capacity	56.8	55.5	54.9	55.8	54.6	54.3	47.7	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
Sequence #	1	1	1	1	1	1	1	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
150 hp compressor																							

B 1: AIRMaster+ Automatic Sequencer Results Page 1



Energy Efficiency Measure Parameters Report
Results (Cont.)

April 2, 2014
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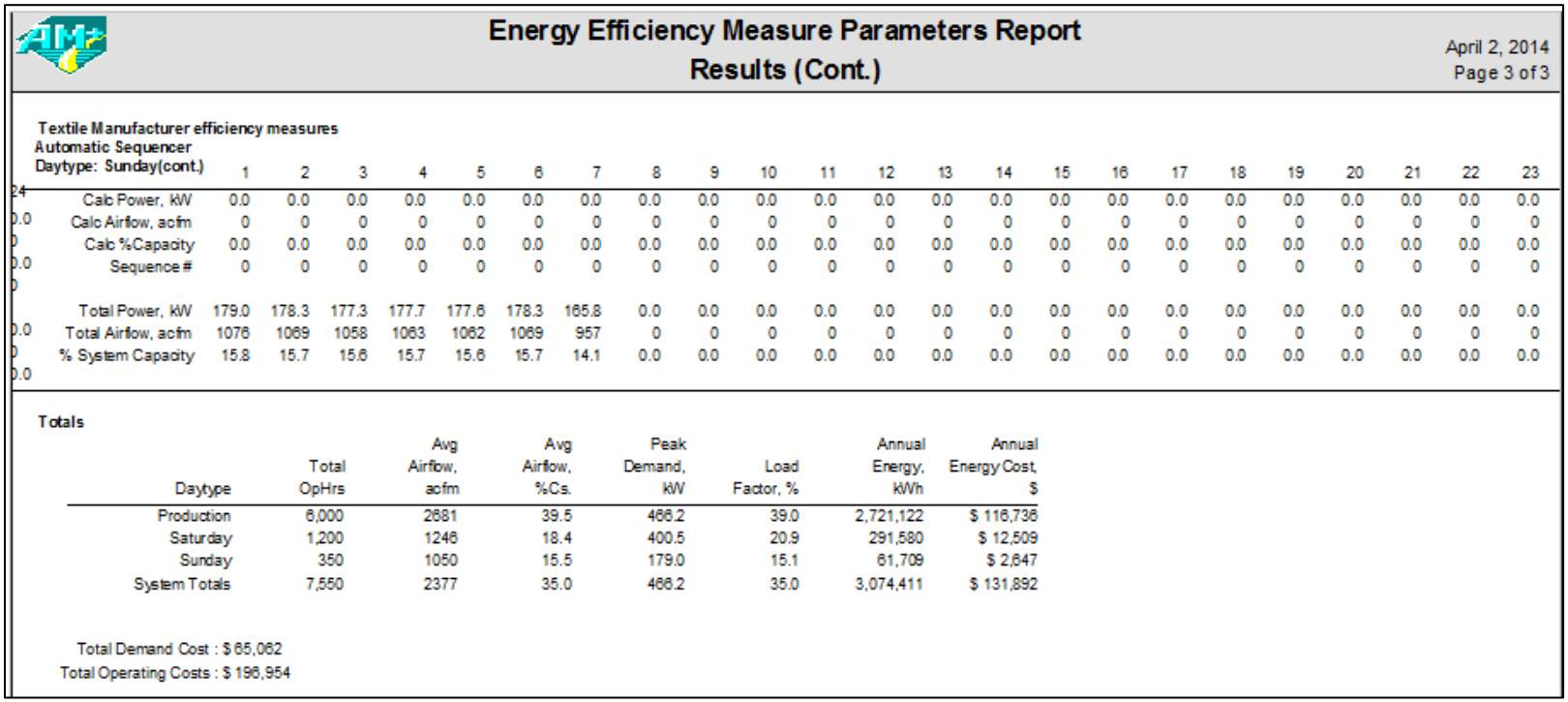
Textile Manufacturer efficiency measures
Automatic Sequencer
Daytype: Saturday(cont.)

	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23
Calc Power, kW	0.0	0.0	0.0	0.0	0.0	0.0	0.0	113.7	113.7	113.7	113.7	113.7	113.7	113.7	113.7	113.7	113.7	113.7	113.7	113.7	113.7	113.7	113.7
Calc Airflow, acfm	0	0	0	0	0	0	0	754	754	754	754	754	754	754	754	754	754	754	754	754	754	754	754
Calc %Capacity	0.0	0.0	0.0	0.0	0.0	0.0	0.0	100.0	100.0	100.0	100.0	100.0	100.0	100.0	100.0	100.0	100.0	100.0	100.0	100.0	100.0	100.0	100.0
Sequence #	0	0	0	0	0	0	0	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1
100 hp compressor																							
Calc Power, kW	0.0	0.0	0.0	0.0	0.0	0.0	0.0	66.1	65.0	64.1	65.8	64.9	66.2	67.3	67.3	66.9	67.0	67.0	64.8	68.1	68.0	67.3	63.2
Calc Airflow, acfm	0	0	0	0	0	0	0	330	319	309	327	317	332	343	343	338	339	340	316	351	351	343	300
Calc %Capacity	0.0	0.0	0.0	0.0	0.0	0.0	0.0	65.5	63.2	61.3	64.9	62.9	65.8	68.1	67.9	67.1	67.3	67.4	62.7	69.7	69.5	68.0	59.5
Sequence #	0	0	0	0	0	0	0	2	2	2	2	2	2	2	2	2	2	2	2	2	2	2	2
500 hp compressor																							
Calc Power, kW	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
Calc Airflow, acfm	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
Calc %Capacity	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
Sequence #	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
Total Power, kW	400.5	398.3	397.3	398.9	396.9	396.4	385.4	179.8	178.7	177.8	179.5	178.6	179.9	181.0	181.0	180.6	180.6	180.7	178.5	181.7	181.7	181.0	176.9
Total Airflow, acfm	1712	1672	1655	1683	1647	1638	1438	1084	1073	1063	1081	1071	1086	1097	1097	1092	1093	1094	1070	1105	1104	1097	1054
% System Capacity	25.2	24.6	24.4	24.8	24.3	24.1	21.2	16.0	15.8	15.7	15.9	15.8	16.0	16.2	16.2	16.1	16.1	16.1	15.8	16.3	16.3	16.2	15.5

Daytype: Sunday

	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23
600 hp compressor																							
Calc Power, kW	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
Calc Airflow, acfm	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
Calc %Capacity	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
Sequence #	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
150 hp compressor																							
Calc Power, kW	113.7	113.7	113.7	113.7	113.7	113.7	113.7	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
Calc Airflow, acfm	754	754	754	754	754	754	754	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
Calc %Capacity	100.0	100.0	100.0	100.0	100.0	100.0	100.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
Sequence #	1	1	1	1	1	1	1	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
100 hp compressor																							
Calc Power, kW	65.3	64.6	63.6	64.0	64.0	64.6	52.2	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
Calc Airflow, acfm	322	315	304	309	308	315	203	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
Calc %Capacity	63.8	62.4	60.3	61.2	61.0	62.4	40.2	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
Sequence #	2	2	2	2	2	2	2	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
500 hp compressor																							

B 2: AIRMaster+ Automatic Sequencer Results Page 2



Energy Efficiency Measure Parameters Report
Results (Cont.)

April 2, 2014
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Textile Manufacturer efficiency measures
Automatic Sequencer
Daytype: Sunday(cont.)

	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23
Calc Power, kW	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
Calc Airflow, acfm	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
Calc %Capacity	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
Sequence #	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
Total Power, kW	179.0	178.3	177.3	177.7	177.6	178.3	165.8	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
Total Airflow, acfm	1076	1069	1058	1063	1062	1069	957	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
% System Capacity	15.8	15.7	15.6	15.7	15.6	15.7	14.1	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0

Totals

Daytype	Total OpHrs	Avg Airflow, acfm	Avg Airflow, %Cs.	Peak Demand, kW	Load Factor, %	Annual Energy, kWh	Annual Energy Cost, $
Production	6,000	2681	39.5	466.2	39.0	2,721,122	$ 116,736
Saturday	1,200	1246	18.4	400.5	20.9	291,580	$ 12,509
Sunday	350	1050	15.5	179.0	15.1	61,709	$ 2,647
System Totals	7,550	2377	35.0	466.2	35.0	3,074,411	$ 131,892

Total Demand Cost : $ 65,062
Total Operating Costs : $ 196,954

B 3: AIRMaster+ Automatic Sequencer Results Page 3

Appendix C-AIRMaster+ Reduce Air Leaks Inputs

Energy Efficiency Measure Parameters Report
Reduce Air Leaks

AIRMaster+
April 2, 2014
Page 1 of 1

For: Company Name
By: Your Name

EEM Scenario : Textile Manufacturer efficiency me
Facility : Textile Manufacturer
System : Textile Manufacturer
Measured data : Airflow, %capacity

Description : Reduce Air Leaks
Measure cost : $ 800

4,000

Compressor Operations to Feed Leaks

Compressor	Units	Airflow
600 hp compressor	%Cap	0.0
150 hp compressor	%Cap	0.0
100 hp compressor	%Cap	50.0
500 hp compressor	%Cap	0.0

Lowest hourly system airflow, acfm : 3103

Leak Airflow Values

	Leaks, acfm	% System Total
Peak system requirement + Leaks	3103	46.3
Leaks	251	3.7
Peak system requirement	2852	42.6
Reduce leaks by 251 acfm		100.0 %

C 1: AIRMaster+ Reduce Air Leaks Inputs

Appendix D-AIRMaster+ Savings Summary

Energy Efficiency Measure Parameters Report
Savings Summary

AIRMaster+
April 7, 2014
Page 1 of 1

For: Company Name
By: Your Name

Facility : Textile Manufacturer
System : Textile Manufacturer
EEM Scenario : Textile Manufacturer efficiency measures

Savings Summary

Description	Energy Savings, kWh	Energy Savings, $	Energy Savings, %	Demand Savings, kW	Demand Savings, $	Installed Cost, $	Total Savings, $	Simple Payback, years
Automatic Sequencer	675,438	28,976	18.0	114.3	15,950	10,000	44,926	0.2
Reduce Air Leaks	128,998	5,534	3.4	13.7	1,917	4,000	7,451	0.5
TOTALS	804,436	34,510	21.5	128.0	17,866	14,000	52,377	0.3

D 1: AIRMaster+ Energy Efficiency Measures Savings Summary